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RESEARCH MEMORANDUM

for the

Bureau of Ordnance, Navy Department

INVESTIGATION OF TURBINE OF MARK 25 TORPEDO POWER PLANT WITH FIVE NOZZLE DESIGNS

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INVESTIGATION OF TURBINE OF MARK 25 TORPEDO POWER PLANT WITH FIVE NOZZLE DESIGNS

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SUMMARY

Efficiency investigations were made on the two-stage turbine from a Mark 25 aerial torpedo to determine the performance of the unit with five different turbine nozzles. The output of the turbine blades was computed by analyzing the windage and mechanical-friction losses of the unit. A method was developed for measuring the change in turbine clearances with changed operating conditions. The turbine was found to be most efficient with a cast nozzle having a sharp-edged inlet to the nine nozzle ports.

INTRODUCTION

Torpedoes operating on a combustion cycle require a high-pressure-ratio gas turbine to drive the propellers of the unit. Some types of rocket may employ similar turbines to operate fuel pumps. Both applications involve the extraction of maximum power over a short time with minimum size and weight of the power plant and the fuel load. Because of the need for information on turbines of this type, at the request of the Bureau of Ordnance, Navy Department, an investigation is being conducted at the NACA Cleveland laboratory of a two-stage turbine from an aerial torpedo to determine the performance of the unit under steady-state conditions with various types of turbine nozzle and to compare the effect of the nozzle designs on the over-all turbine efficiency.

A series of five turbine nozzles was investigated to determine the effect of pressure ratio, blade-jet speed ratio, and nozzle design on turbine efficiency. This investigation covered a range of pressure ratios from 8 to 20 and turbine speeds from

6000 to 18,000 rpm with an inlet-gas temperature of 1000° F and an inlet-gas pressure of 95 pounds per square inch gage. The true output of the nozzle-and-turbine-blade combinations was determined by evaluating the power losses of the turbine due to windage and mechanical friction. A method was developed for measuring the change in turbine clearances with changed turbine operating conditions, and the effect of nozzle-wheel clearance on turbine efficiency was investigated.

APPARATUS AND INSTRUMENTATION

Turbine. - The Mark 25 torpedo power plant is a two-stage counterrotating impulse turbine with integral speed-reduction and power-equalizing gearing. The power-output shafts are also counterrotating and operate at 0.099 turbine speed. A combining gearbox was specially designed for this installation to convert the dual rotation of the torpedo power-output shafts into single-direction rotation.

A sketch of the nozzle-and-turbine-wheel assembly is shown in figure 1. The gas expands through the partial-admission nozzles and enters the forward turbine blades at supersonic relative velocity. Because considerable gas-velocity energy is absorbed by the forward turbine, the flow leaving the first rotor and entering the counterrotating rear wheel is subsonic. Although a small amount of reaction is employed in the rear turbine to extract maximum energy, the rear turbine is included in this design mainly to eliminate gyroscopic effects. The shroud-band diameters of the forward and rear turbine wheels are 11.000 and 11.320 inches, respectively.

Nozzles. - The high inlet-gas temperatures and pressures, the high pressure ratios, and the low gas weight flows for this turbine necessitate special care in the design and the construction of the turbine nozzles. The high pressure ratios in the nozzles cause supersonic flow, and very small nozzle throats and partial admission are necessary for low weight flow.

The nine-port nozzles (A, E, G, and H) have 90° nozzle-arc gas admission, whereas the three-port nozzle F has 30° nozzle-arc gas admission. Nozzles A and E are part of a series of nozzles investigated under actual torpedo conditions at Massachusetts Institute of Technology (reference 1). Nozzles F, G, and H are a continuation of this nozzle series.

Nozzle A has rounded inlets to the nine rectangular, converging-diverging nozzle ports; the nine nozzle throats are approximately 0.25 by 0.10 inch. The nozzle was produced by the precision lost-wax casting method and has hand-filed nozzle ports.

The difficulty in manufacturing nozzles similar to nozzle A led to the design of a nozzle that could be more easily fabricated. Nozzle E (fig. 2(a)) has reamed ports with rounded inlets. The ports are cylindrical with no area divergence; the necessary expansion takes place in the axial-clearance space between the nozzle and the forward turbine buckets. The nine nozzle ports are approximately 0.163 inch in diameter. Nozzle F is similar to nozzle E except that the six ports nearest the gas inlet are blocked by a welded plate, leaving only three ports in use. The nozzle ports are also 0.163 inch in diameter.

In order to provide a more favorable flow profile and to prevent the combustion gas from spilling above or below the forward-wheel blades, the shrouded nine-port nozzle G (fig. 2(b)) was designed with the same nozzle dimensions as nozzle E but with a shroud projecting axially 0.216 inch from the outlet face.

Nozzle H (fig. 2(c)) was produced by refinement of both design and casting technique to give more accurate dimensional control and a smoother nozzle surface. This nozzle has nine rectangular ports with sharp inlet edges and throat sizes of approximately 0.25 by 0.10 inch.

Setup and instrumentation. - The setup used for this investigation is shown in figure 3. A 300-horsepower electric dynamometer was used to absorb the turbine output and to drive the turbine during motoring runs. The dynamometer speed was measured with a chronometric tachometer. Dynamometer torque measurements were made with a scale having a range of 0 to 250 pounds.

Combustion gases at a pressure of 95 pounds per square inch gage were furnished by a burner using unleaded gasoline and air. The altitude exhaust at the turbine outlet was used to obtain the desired range of pressure ratios across the turbine. The air flow was metered by a standard A.S.M.E. orifice in the inlet-air pipe. The fuel flow was measured by means of a calibrated rotameter.

Gas-stream temperatures were measured by triple-shielded thermocouples of 0.50-inch outside diameter and 0.75 inch long. The inlet-gas temperatures were taken by two thermocouples mounted in the insulated inlet-gas pipe. Two thermocouples were located in the exhaust cone 2 inches behind the rear turbine wheel. Seven

static-pressure taps were placed on the outer heat-shield ring to provide a pressure survey along the nozzle arc above the wheels in order to evaluate the gas density around the turbine. Static and total pressures were taken in the inlet- and outlet-gas pipes. In addition, a total-pressure tube insensitive to yaw was mounted in the exhaust cone 2 inches behind the rear turbine wheel in the same plane as the exhaust thermocouple.

The turbine was modified for long-duration efficiency runs by removing the integral oil pump on the unit and installing an external oil system with instruments to measure the oil temperature and the flow rate. Thermocouples were installed on the four high-speed bearings to guard against failures by overheating of the bearings. Thermocouples were also placed on the front and rear oil seals nearest the turbine wheels to indicate any hot-gas leakage past the seals into the turbine bearings. Sufficient temperature measurements were taken on the aluminum housing and cover of the unit to avoid operation above the safe-temperature limits of these parts. Thermocouples were located on the stainless-steel heat shields to indicate the operating temperatures.

During runs to determine the change in axial nozzle-wheel clearance and radial nozzle setting with operating conditions, clearance indicators were installed on the turbine as shown in figure 4. The method of operation of the axial-clearance indicators is illustrated by figure 5. The gage at the left is shown resting on a lug welded to the turbine nozzle. When the tip of the gage is rotated 180° by turning the handle to the position shown for the gage at the right, the tip is allowed to move past the lug and touch the turbine wheel. The difference between the dial-indicator readings with the indicator tip in the two positions is a measure of the clearance.

<u>Precision.</u> - The precision of the measurements was within the following limits:

Air flow, percent	±1.00
Torque, foot-pounds	±0.15
Speed, rpm	±5
Inlet-gas pressure, percent	±0.50
Pressure, inches mercury	±0.05

PROCEDURE

The investigation of the Mark 25 torpedo power plant was divided into the following phases:

(1) Power runs to determine turbine efficiency and over-all performance

(2) Motoring runs to evaluate the windage and mechanical losses of the turbine unit for use in efficiency calculations

(3) Clearance studies to find the true running clearance between nozzles and turbine wheel

Power runs. - The over-all efficiency of the turbine power plant and combining gear was determined from the power output with the various nozzles at several pressure ratios and turbine speeds over the range of operation. With the inlet-gas conditions maintained at 95 pounds per square inch gage and 1000° F, the turbine speed was varied from 6000 to 18,000 rpm at pressure ratios of 8, 10, 15, and 20. (Because of air-flow limitations, the maximum pressure ratio obtained with nozzle H was 19.) The values of power output obtained during these runs represent the output of the turbine wheels with the windage and the gear and bearing friction in the power plant subtracted.

Motoring runs. - The rotation losses of the turbine unit due to gears, bearings, and turbine disks were obtained by motoring the unit with disks instead of the turbine wheels. The forward and rear disks were 10.075 and 9.955 inches in diameter, respectively, or exactly the same diameter as the root diameters of the turbine blades. The power required to motor these disks at speeds from 6000 to 18,000 rpm and at different air densities in the turbine case represents the total power absorbed through mechanical losses in the gears and the bearings and through windage of the turbine disks.

The motoring runs were then repeated with standard turbine wheels with forward- and rear-wheel shroud-band diameters of 11.000 and 11.320 inches, respectively. The power then required represents all the losses of the disk unit plus the air-pumping effect of the turbine blades. All motoring runs were made without air flow through the turbine, the turbine nozzle being blocked. Air densities in the turbine case were varied by varying the turbine-outlet-gas pressure.

Clearance studies. - Runs were made at an inlet-gas pressure of 95 pounds per square inch gage, temperatures from 500° to 1500° F, and pressure ratios of 10, 15, and 20 at speeds of 6000 and 12,000 rpm with indicators installed to measure the changes in radial nozzle setting and axial nozzle-turbine clearance with operating conditions. The thermal changes in radial setting and axial clearance were determined by operating the turbine at low speeds with varying inlet-gas temperature. In order to indicate

the change in radial wheel dimensions with centrifugal force, the turbine was motored at speeds of 6000 to 18,000 rpm with sufficient cold air allowed to enter the nozzle inlet to keep the wheels, nozzle box, and turbine case at approximately room temperature so that thermal effects would be eliminated.

Radial setting of the nozzle ports for turbine-efficiency runs with nozzles A, E, F, and G were made to conform to the recommendations of reference 1. Accordingly, a radial setting of 5.283 inches from the turbine-shaft center line to the midpoint of the nozzle port was used for each of these nozzles. Nozzle H, because of its different port design and larger flow area, required experimentation to obtain the best radial setting for the final turbine-efficiency runs. The proper value of 5.258 inches was obtained by over-all turbine-efficiency runs over a range of radial settings from 5.233 to 5.308 inches from turbine-shaft center line to average midpoint of nozzle ports.

The effect of axial clearance on turbine efficiency was investigated by varying the axial nozzle-wheel clearance for nozzle A from 0.030 to 0.090 inch and determining the over-all turbine efficiency for each axial-clearance setting.

Axial nozzle-wheel clearances for turbine-efficiency runs were set, on the basis of the clearance-indicator studies, to give 0.030-inch clearance when the unit reached operating temperature. Because the turbine blades are set back from the edge of the turbine wheel 0.025 inch, the axial clearance from nozzle to blade was thus 0.055 inch.

CALCULATIONS

The isentropic enthalpy drop available for an expansion from the turbine-inlet-gas total temperature and pressure to the outlet-gas static pressure was computed from the air tables of reference 2 and corrected for the effect of the fuel input.

The blade-jet speed ratio is the ratio of the blade velocity at the pitch diameter of the first wheel to the ideal turbine-nozzle jet velocity corresponding to an isentropic expansion from the inlet-gas total temperature and pressure to the turbine-outlet-gas static pressure.

Pressure ratio is the ratio of the inlet-gas total pressure to the outlet-gas static pressure.

Brake efficiency is the ratio of the brake power calculated from the torque and speed of the dynamometer shaft to the isentropic power available; that is

$$\text{brake efficiency} = \frac{\text{brake power}}{\text{isentropic power available}}$$

The mechanical losses of the unit were determined by extrapolating the windage-loss curves to zero air density. The horsepower at zero air density represents the power required to drive the gears and overcome bearing friction and is the mechanical loss for the turbine and the combining gear. The mechanical-loss power added to the brake power gives the power output of the turbine wheels at the shaft; and

$$\text{wheel efficiency} = \frac{\text{brake power + mechanical losses}}{\text{isentropic power available}}$$

The windage losses were obtained from the windage chart for the nozzle arc of admission under consideration.

The power required to motor the unit with disks installed instead of turbine wheels represents the total power absorbed through mechanical losses in the gears and the bearings plus the windage of the disks. These losses are presented in figure 6(a) and table I. Curves for constant disk speed have been extrapolated to zero air density to separate the mechanical losses from the power losses due to the disk windage. The windage and mechanical losses obtained by motoring the standard turbine wheels are presented in figure 6(b) and table II. Slight differences in mechanical losses in the units shown in figures 6(a) and 6(b) can be ascribed to inherent differences in assembly and to curve extrapolation.

The windage loss for the full blade periphery was calculated by subtracting the disk-windage loss determined from figure 6(a) from the disk-and-blade-windage loss obtained from figure 6(b). Blade-windage loss derived in this manner is considered the loss of the turbine blades with 0° nozzle-arc gas admission.

When the nine-port turbine nozzles with 90° nozzle-arc gas admission (nozzles A, E, G, and H) are used, 90° of the turbine-blade periphery, or one-fourth of the turbine buckets, is active and hence is not subject to windage loss. The windage losses for the unit with these nozzles are composed of the total disk

windage plus three-fourths of the windage due to the turbine blades. Accordingly, figure 7(a) was prepared from the total disk windage plus three-fourths of the additional windage loss due to the turbine blades. This chart can be used to find directly the windage and mechanical losses of the unit with the nine-port nozzles if the gas density in the turbine case is known.

Nozzle F has three ports with 30° nozzle-arc gas admission; hence 30° of the turbine-bucket periphery, or one-twelfth of the turbine blades, is active and thus has no windage losses. Figure 7(b) was prepared by using the total disk windage plus eleven-twelfths of the additional windage loss due to the turbine blades.

The windage-loss power added to the mechanical-loss power and the brake power gives the work output at the turbine blades, and

$$\text{blade efficiency} = \frac{\text{brake power + mechanical losses + windage}}{\text{isentropic power available}}$$

RESULTS

Turbine efficiency. - The individual performance of the turbine at an inlet-gas pressure of 95 pounds per square inch gage and temperature of 1000° F and pressure ratios of 8 to 20 with nozzles A, E, F, G, and H are shown in figures 8 to 12 and tables III to VII, respectively. All data shown in figures 8 to 12 were taken with 0.030-inch axial nozzle-wheel running clearance. The maximum brake efficiency of 0.53 was obtained with nozzle H at a blade-jet speed ratio of approximately 0.21 for a pressure ratio of 8 (fig. 12(a)). At the maximum brake-efficiency point, the wheel efficiency (which credits the turbine with the work necessary to drive the gears and bearings) was approximately 0.56. If the turbine is also credited with the work necessary to overcome windage losses, the efficiency was 0.58 at the foregoing conditions. Nozzle H also showed the highest blade efficiency (0.64) of the nine-port nozzles at a blade-jet speed ratio of 0.295 and pressure ratio of 8 (fig. 12(a)) although the peak blade efficiency (which would be at a higher blade-jet speed ratio) could not be determined because of the 18,000-rpm speed limit of the turbine. The higher efficiencies of the turbine with nozzle H is evidence of either (a) lower losses in the nozzle, or (b) more favorable matching of turbine flow area with the greater flow area of this nozzle. Nozzle H has, for example, about 20 percent greater gas mass flow than nozzle A.

The difference in the general trends of the curves of brake efficiency and blade efficiency can be ascribed to the

approximately cubic increase of windage losses with speed. The turbine is therefore heavily penalized by the use of partial-admission nozzles as shown by the three-port nozzle F (fig. 10), for which brake efficiency drops off sharply with increased blade-jet speed ratio but blade efficiency increases with increased speed.

The reamed nozzles generally showed lower efficiencies than the cast nozzles. The addition of a shroud to provide a guide for the nozzle jets resulted in a marked decrease in efficiency (fig. 11), presumably because of increased shock and eddy losses.

The effect of pressure ratio on brake efficiency with nozzle H at an inlet-gas temperature of 1000° F is shown in figure 13(a). The brake efficiency varies only slightly with pressure ratio with this nozzle; over the range of pressure ratios from 8 to 19, the maximum difference in efficiency was 0.01. The effect of inlet-gas temperature on brake efficiency with nozzle A is shown in figure 13(b). An increase in inlet-gas temperature from 500° to 1000° F resulted in an efficiency decrease of 0.03 owing to the increased heat loss from the unit.

Clearances. - The results of clearance-indicator studies of the changes in radial setting and axial clearance between the turbine nozzle and the turbine wheel with changed operating conditions are shown in figure 14 and table VIII. Although the temperatures in the turbine case are affected by turbine speed and pressure ratio as well as the inlet-gas temperature, the changes in setting and clearance were plotted against inlet-gas temperature (which does not reflect those variables) because an average clearance-change value was desired for the turbine operating over a wide range of speed and pressure-ratio conditions for a given inlet-gas temperature. The data plotted in this manner yield a range through which the clearance may vary with a given inlet-gas temperature.

From this chart, a correction of the clearance setting of the turbine may be made before operation to obtain the desired value of running clearance when the parts reach operating temperature. The maximum change in radial setting over the range of inlet-gas temperatures was a decrease of 0.013 inch. The difference between the left and right radial indicators is evidence of a slight tilting of the nozzle. Axial changes in clearance were slightly more pronounced, the maximum decrease being about 0.017 inch. A radial-setting decrease of about 0.005 inch and an axial-clearance decrease of about 0.015 inch were chosen as safe allowances for changes in clearance over a range of inlet-gas temperatures from room temperature to 1500° F.

The results of motoring runs to determine the expansion of the turbine wheels due to centrifugal force are given in table IX. Although the runs were made with room temperature in the nozzle casing, rough calculations indicate that turbine operating temperatures would have only a small effect on radial expansion due to centrifugal force for this turbine. At a turbine speed of about 18,000 rpm with room temperature, the radial expansion was found to be 0.0045 inch.

The variation of brake efficiency with radial setting of nozzle H is shown in figure 15(a). As the radial setting of the nozzle was varied from 5.308 to 5.233 inches, the turbine efficiency increased slightly to a maximum value at a radial setting of 5.258 inches then, with a further decrease in setting, dropped sharply.

The variation of brake efficiency with axial nozzle-wheel clearance of nozzles A and E is shown in figure 15(b). The brake efficiency of nozzle A improved as the running clearance was decreased from 0.090 to 0.030 inch; at 0.030-inch axial nozzle-wheel clearance, the turbine is less efficient with nozzle E than with nozzle A. At a running clearance of 0.090 inch, however, the turbine efficiency with nozzle E is slightly improved over that with nozzle A, which indicates that adequate clearance space must be provided for free gas-jet expansion with a nondivergent nozzle for greatest turbine efficiency.

As a result of the clearance-indicator studies, the axial nozzle-wheel clearance and the radial setting were positioned at 0.015 and 0.005 inch, respectively, more than the running clearances desired during turbine operation.

SUMMARY OF RESULTS

The mechanical and windage losses of a Mark 25 aerial-torpedo power plant were determined and the efficiency of the turbine, based on these loss determinations, was investigated for five turbine nozzles. Clearances were also studied. The following results were obtained:

1. The turbine was found to be most efficient with a cast, sharp-edged-inlet, nine-port nozzle with 90° nozzle-arc gas admission.

2. The maximum brake efficiency of the turbine with the cast, sharp-edged-inlet nozzle at an inlet pressure of 95 pounds per

square inch gage, an inlet-gas temperature of 1000° F, a pressure ratio of 8, and a blade-jet speed ratio of 0.21, was 0.53. If the turbine is credited with the work necessary to drive the gears and the bearings, the efficiency at the foregoing conditions was 0.56. If the turbine is also credited with the work necessary to overcome windage losses, the efficiency was 0.58.

3. Windage and mechanical-loss charts that have been prepared gave the operating loss in horsepower for various turbine-casing pressures, speeds, and nozzle arcs of 30° and 90°.

4. The variation of radial nozzle setting and axial nozzle-wheel clearance with turbine-inlet-gas temperature was small. A radial-setting decrease of about 0.005 inch and an axial-clearance decrease of about 0.015 inch were chosen as safe allowances for changes in clearance over a range of inlet-gas temperatures from room temperature to 1500° F.

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TABLE I - SUMMARY OF DATA FOR WINDAGE AND MECHANICAL LOSSES

FOR MARK 25 TURBINE DISKS

Turbine speed (rpm)	Horsepower to drive turbine	Air temperature in turbine case (°F)	Pressure in turbine case (in. Hg abs.)
6,069	0.80	134	29.36
	.80	124	24.36
	.76	129	19.36
	.74	132	14.36
	.70	135	9.36
8,092	1.33	158	29.36
	1.28	159	24.36
	1.20	165	19.36
	1.15	164	14.36
	1.09	163	9.36
10,115	2.13	187	29.36
	2.07	183	24.39
	1.97	196	19.38
	1.87	194	14.40
	1.77	187	9.36
12,138	2.96	239	29.36
	2.68	245	24.40
	2.60	247	19.36
	2.48	241	14.37
	2.36	214	9.41
14,161	4.01	265	29.36
	3.64	275	24.42
	3.41	286	19.36
	3.27	270	14.44
	3.08	229	9.99
16,184	5.17	304	29.36
	4.69	341	24.44
	4.37	343	19.43
	4.05	330	14.46
	3.89	281	10.81
18,207	6.48	379	29.36
	5.82	392	24.38
	5.46	389	19.40
	5.04	356	14.42
	4.68	315	10.42

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TABLE II - SUMMARY OF DATA FOR WINDAGE AND MECHANICAL LOSSES

FOR MARK 25 TURBINE DISKS AND BLADES

Turbine speed (rpm)	Horsepower to drive turbine	Air temperature in turbine case (°F)	Pressure in turbine case (in. Hg abs.)
6,069	1.34	182	29.30
	1.26	243	25.31
	1.20	206	18.73
	1.08	163	14.32
	.94	147	9.66
8,092	2.32	230	29.32
	2.21	237	25.45
	1.97	224	19.35
	1.83	197	14.71
	1.63	179	9.58
10,115	3.66	306	29.36
	3.33	297	24.11
	2.93	304	19.19
	2.70	299	15.26
	2.33	281	9.76
12,138	5.53	369	29.42
	5.00	351	23.79
	4.44	369	20.15
	3.84	364	15.56
	3.20	339	9.69
14,161	7.67	461	29.46
	6.95	461	24.90
	5.83	462	18.58
	4.90	443	13.70
	4.11	416	9.41
16,184	10.46	558	29.54
	9.65	463	23.60
	8.21	498	18.86
	6.99	481	14.02
	6.03	465	10.58
18,207	15.83	647	29.63
	11.70	459	15.28
	9.72	432	11.20
	8.10	383	7.99

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TABLE III - SUMMARY OF EFFICIENCY DATA FOR MARK 25 TURBINE

WITH NOZZLE A

[Inlet-gas temperature, 1000° F; inlet-gas pressure, 95 lb/sq in. gage]

Pressure ratio	Air mass flow (lb/hr)	Fuel-air ratio	Horsepower available from isentropic expansion	Turbine speed (rpm)	Brake horsepower	Blade-jet speed ratio	Gas density in turbine case (lb/cu ft)
8	955.2	0.0135	61.09	6,089	20.97	0.0987	0.0354
	955.9	.0135	61.14	8,072	24.63	.1308	.0358
	955.9	.0135	61.14	10,125	27.16	.1641	.0362
	955.9	.0135	61.14	12,158	28.29	.1970	.0369
	956.6	.0135	61.18	14,161	28.23	.2295	.0370
	955.9	.0135	61.14	16,164	27.11	.2620	.0369
	956.6	.0135	61.18	18,207	23.94	.2951	.0367
10	960.3	0.0135	66.20	6,069	23.56	0.0948	0.0294
	960.3	.0135	66.20	8,122	27.97	.1268	.0301
	958.2	.0135	66.06	10,115	30.67	.1580	.0302
	959.3	.0135	66.12	12,138	32.12	.1895	.0304
	958.2	.0135	66.06	14,161	32.11	.2211	.0304
	958.2	.0135	66.06	16,194	31.06	.2529	.0305
	958.2	.0135	66.06	18,227	28.41	.2846	.0303
15	958.2	0.0135	73.98	6,069	25.06	0.0896	0.0210
	958.2	.0135	73.98	8,092	30.00	.1195	.0210
	958.2	.0135	73.98	10,125	33.57	.1495	.0213
	960.1	.0135	74.13	12,138	35.80	.1792	.0209
	958.2	.0135	73.98	14,161	36.40	.2091	.0211
	958.2	.0135	73.98	16,194	35.70	.2391	.0215
	958.2	.0135	73.98	18,187	33.98	.2685	.0220
20	958.2	0.0135	79.10	6,059	25.42	0.0870	0.0167
	959.3	.0135	79.18	8,092	30.85	.1162	.0169
	959.3	.0135	79.18	10,125	34.87	.1454	.0167
	958.2	.0135	79.10	12,148	37.43	.1745	.0166
	958.2	.0135	79.10	14,151	38.47	.2032	.0165
	958.2	.0135	79.10	16,214	37.94	.2328	.0169
	958.2	.0135	79.10	18,187	36.14	.2612	.0175

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TABLE IV - SUMMARY OF EFFICIENCY DATA FOR MARK 25 TURBINE

WITH NOZZLE E

[Inlet-gas temperature, 1000° F; inlet-gas pressure, 95 lb/sq in. gage]

Pressure ratio	Air mass flow (lb/hr)	Fuel-air ratio	Horsepower available from isentropic expansion	Turbine speed (rpm)	Brake horsepower	Blade-jet speed ratio	Gas density in turbine case (lb/cu ft)
8	976.5	0.0138	62.49	6,099	20.96	0.0988	0.0338
	977.6	.0138	62.56	8,112	24.01	.1315	.0345
	977.6	.0138	62.56	10,125	25.53	.1641	.0348
	976.5	.0138	62.49	12,148	25.66	.1969	.0358
	976.5	.0138	62.49	14,141	27.77	.2292	.0360
	977.6	.0138	62.56	16,164	26.26	.2620	.0360
	976.5	.0138	62.49	18,207	23.52	.2951	.0357
10	977.6	0.0138	67.42	6,079	23.00	0.0949	0.0280
	978.6	.0138	67.48	8,112	27.24	.1267	.0282
	977.6	.0138	67.42	10,105	29.94	.1578	.0282
	977.6	.0138	67.42	12,138	31.04	.1895	.0287
	977.6	.0138	67.42	14,151	30.87	.2210	.0291
	977.6	.0138	67.42	16,204	29.69	.2530	.0291
	977.6	.0138	67.42	18,197	27.52	.2842	.0290
15	977.4	0.0138	75.50	6,079	24.22	0.0897	0.0199
	976.3	.0138	75.41	8,132	29.08	.1200	.0202
	977.4	.0138	75.50	10,125	32.20	.1495	.0202
	976.3	.0138	75.41	12,168	33.60	.1796	.0201
	974.6	.0138	75.27	14,181	34.02	.2093	.0201
	976.3	.0138	75.41	16,194	35.01	.2391	.0205
	974.7	.0138	75.28	18,227	33.16	.2691	.0207
20	975.6	0.0138	80.56	6,059	25.58	0.0865	0.0162
	975.6	.0138	80.56	8,102	31.16	.1157	.0163
	975.6	.0138	80.56	10,115	34.97	.1445	.0165
	976.5	.0138	80.63	12,128	37.49	.1732	.0161
	976.5	.0138	80.63	14,171	38.43	.2024	.0163
	976.5	.0138	80.63	16,204	37.91	.2314	.0163
	976.5	.0138	80.63	18,217	36.62	.2601	.0168

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TABLE V - SUMMARY OF EFFICIENCY DATA FOR MARK 25 TURBINE

WITH NOZZLE F

[Inlet-gas temperature, 1000° F; inlet-gas pressure, 95 lb/sq in. gage]

Pressure ratio	Air mass flow (lb/hr)	Fuel-air ratio	Horsepower available from isentropic expansion	Turbine speed (rpm)	Brake horsepower	Blade-jet speed ratio	Gas density in turbine case (lb/cu ft)
8	247.3	0.0208	15.99	6,059	6.29	0.0982	0.0354
	247.3	.0208	15.99	8,112	6.74	.1315	.0358
	247.3	.0208	15.99	10,125	6.54	.1641	.0355
	247.3	.0212	16.00	12,148	5.24	.1969	.0356
	247.3	.0212	16.00	14,141	3.63	.2292	.0348
	247.1	.0212	15.98	16,184	1.76	.2623	.0338
	247.3	.0212	16.00	18,217	-.48	.2952	.0324
10	246.6	0.0213	17.20	6,069	6.54	0.0948	0.0292
	246.6	.0213	17.20	8,102	7.34	.1265	.0288
	246.6	.0213	17.20	10,115	7.40	.1580	.0284
	246.6	.0213	17.20	12,138	6.48	.1895	.0278
	246.6	.0213	17.20	14,171	5.09	.2213	.0275
	246.4	.0213	17.18	16,194	3.36	.2529	.0273
	246.6	.0213	17.20	18,207	1.20	.2843	.0266
15	247.9	0.0212	19.37	6,079	6.51	0.0897	0.0209
	247.9	.0212	19.37	8,082	7.48	.1193	.0206
	247.9	.0212	19.37	10,115	7.83	.1493	.0205
	248.1	.0212	19.38	12,138	7.76	.1792	.0204
	248.1	.0212	19.38	14,161	7.00	.2091	.0203
	248.1	.0212	19.38	16,184	5.49	.2389	.0201
	248.1	.0212	19.38	18,197	3.60	.2686	.0202
20	248.1	0.0212	20.73	6,089	6.32	0.0870	0.0164
	248.1	.0212	20.73	8,112	7.51	.1158	.0158
	248.1	.0212	20.73	10,115	8.37	.1444	.0153
	248.1	.0212	20.73	12,128	8.51	.1732	.0153
	248.3	.0211	20.75	14,161	7.89	.2022	.0153
	248.1	.0212	20.73	16,184	6.93	.2311	.0154
	248.3	.0211	20.75	18,207	5.40	.2600	.0154

TABLE VI - SUMMARY OF EFFICIENCY DATA FOR MARK 25 TURBINE

NOZZLE G

[Inlet-gas temperature, 1000° F; inlet-gas pressure, 95 lb/sq in. gage]

820

Pressure ratio	Air mass flow (lb/hr)	Fuel-air ratio	Horsepower available from isentropic expansion	Turbine speed (rpm)	Brake horsepower	Blade-jet speed ratio	Gas density in turbine case (lb/cu ft)
8	977.6	0.0138	62.56	6,039	20.50	0.0979	0.0334
	976.2	.0138	62.47	8,122	23.82	.1316	.0334
	977.6	.0138	62.56	10,115	25.60	.1639	.0332
	977.6	.0138	62.56	12,138	26.08	.1967	.0340
	977.6	.0138	62.56	14,191	25.30	.2300	.0341
	977.6	.0138	62.56	16,204	23.50	.2626	.0342
	977.6	.0138	62.56	18,227	20.42	.2954	.0340
10	977.6	0.0138	67.42	6,069	22.50	0.0948	0.0275
	976.9	.0138	67.37	8,122	26.63	.1268	.0268
	976.9	.0138	67.37	10,115	28.67	.1580	.0266
	976.9	.0138	67.37	12,128	29.38	.1894	.0273
	976.7	.0138	67.36	14,151	28.73	.2210	.0274
	976.0	.0138	67.31	16,194	26.95	.2529	.0276
	977.6	.0138	67.42	18,217	24.25	.2845	.0275
15	976.5	0.0138	75.43	6,069	23.84	0.0896	0.0191
	976.5	.0138	75.43	8,112	28.82	.1197	.0189
	978.1	.0138	75.55	10,125	31.83	.1495	.0189
	977.2	.0138	75.48	12,138	33.24	.1792	.0191
	976.5	.0138	75.43	14,171	32.88	.2092	.0193
	977.2	.0138	75.48	16,174	31.50	.2388	.0196
	977.2	.0138	75.48	18,207	29.34	.2688	.0199
20	976.5	0.0138	80.64	6,069	24.90	0.0867	0.0153
	977.2	.0138	80.70	8,122	30.06	.1160	.0156
	976.5	.0138	80.64	10,125	33.43	.1446	.0154
	977.2	.0138	80.70	12,158	35.26	.1736	.0152
	976.5	.0138	80.64	14,161	35.61	.2022	.0152
	978.1	.0138	80.76	16,204	34.02	.2314	.0158
	977.2	.0138	80.70	18,177	32.23	.2596	.0162

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TABLE VII - SUMMARY OF EFFICIENCY DATA FOR MARK 25 TURBINE

WITH NOZZLE H

[Inlet-gas temperature, 1000° F; inlet-gas pressure, 95 lb/sq in. gage]

Pressure ratio	Air mass flow (lb/hr)	Fuel-air ratio	Horsepower available from isentropic expansion	Turbine speed (rpm)	Brake horsepower	Blade-jet speed ratio	Gas density in turbine case (lb/cu ft)
8	1143.7	0.0143	73.23	6,069	28.24	0.0984	0.0322
	1143.7	.0143	73.23	8,072	33.25	.1308	.0331
	1143.7	.0143	73.23	10,054	36.78	.1630	.0332
	1143.7	.0143	73.23	12,189	38.56	.1976	.0338
	1143.7	.0143	73.23	14,161	38.27	.2295	.0345
	1143.7	.0143	73.23	16,164	36.54	.2620	.0346
	1143.7	.0143	73.23	18,187	33.26	.2948	.0347
10	1152.0	0.0141	79.48	6,069	29.62	0.0948	0.0257
	1152.0	.0141	79.48	8,112	35.50	.1267	.0265
	1152.0	.0141	79.48	10,105	39.49	.1578	.0273
	1152.0	.0141	79.48	12,138	41.00	.1895	.0278
	1144.1	.0142	78.94	14,171	41.10	.2213	.0284
	1144.1	.0142	78.94	16,164	39.63	.2524	.0288
	1152.0	.0141	79.48	18,187	36.80	.2840	.0293
15	1144.1	0.0142	88.43	6,089	31.02	0.0899	0.0198
	1144.1	.0142	88.43	8,122	37.87	.1199	.0197
	1144.1	.0142	88.43	10,135	42.35	.1496	.0201
	1144.1	.0142	88.43	12,138	45.16	.1792	.0200
	1144.1	.0142	88.43	14,171	46.84	.2092	.0204
	1144.1	.0142	88.43	16,184	46.13	.2389	.0214
	1144.1	.0142	88.43	18,217	43.28	.2689	.0224
19	1144.5	0.0142	93.51	6,099	32.00	0.0876	0.0177
	1144.5	.0142	93.51	8,092	38.72	.1162	.0180
	1144.5	.0142	93.51	10,125	43.81	.1454	.0182
	1144.5	.0142	93.51	12,148	47.00	.1745	.0184
	1144.5	.0142	93.51	14,171	48.75	.2035	.0184
	1144.5	.0142	93.51	16,184	48.37	.2324	.0192
	1144.5	.0142	93.51	18,217	46.35	.2616	.0197

TABLE VIII - SUMMARY OF CLEARANCE INDICATOR DATA FOR THERMAL EXPANSION OF MARK 25 TURBINE

[Inlet-gas pressure, 95 lb/sq in. gage]

Pressure ratio	Turbine speed (rpm)	Inlet-gas temperature (°F)	Outlet-gas temperature (°F)	Change in clearance (in.)			
				Axial		Radial	
				Left gage	Right gage	Left gage	Right gage
-----	0	73	88	0	0	0	0
10	6,069	502	281	-.0120	-.0130	-.0040	-.0035
	6,069	748	465	-.0130	-.0145	-.0050	-.0050
	6,069	1003	662	-.0140	-.0160	-.0070	-.0065
	6,069	1250	851	-.0140	-.0165	-.0075	-.0020
	6,069	1485	1048	-.0090	-.0090	-.0090	0
-----	0	70	80	0	0	0	0
15	6,069	495	261	-.0090	-.0070	-.0030	-.0040
	6,069	739	443	-.0150	-.0115	(a)	-.0040
	6,069	993	620	-.0140	-.0130	(a)	-.0035
	6,069	1248	810	-.0140	-.0100	(a)	0
	6,069	1507	1004	-.0110	-.0075	(a)	.0040
-----	0	68	75	0	0	0	0
20	6,069	503	253	-.0080	-.0085	-.0050	-.0030
	6,069	753	438	-.0105	-.0100	-.0080	-.0035
	6,069	1010	634	-.0120	-.0110	-.0100	-.0045
	6,069	1251	817	-.0135	-.0095	-.0110	-.0020
	6,069	1498	987	-.0120	-.0090	-.0130	.0030
-----	0	70	82	0	0	0	0
15	12,138	489	190	-.0060	-.0080	-.0050	-.0035
	12,138	753	374	-.0090	-.0110	-.0075	-.0045
	12,138	998	542	-.0130	-.0135	-.0095	-.0055
	12,138	1256	712	-.0150	-.0145	-.0110	-.0020
	12,138	1505	883	-.0130	-.0145	-.0090	.0010
-----	0	90	90	0	0	0	0
20	12,138	497	205	-.0050	-.0070	-.0030	-.0025
	12,138	750	370	-.0080	-.0085	-.0055	-.0040
	12,138	1012	524	-.0100	-.0110	-.0080	-.0050
	12,138	1255	690	-.0125	-.0125	-.0090	(a)
	12,138	1508	864	-.0150	-.0115	-.0070	(a)

[a]Indicator failed.

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TABLE IX - SUMMARY OF CLEARANCE-INDICATOR DATA FOR CENTRIFUGAL EXPANSION OF MARK 25 TURBINE

[Room temperature and pressure]

Turbine speed (rpm)	Radial expansion	
	Left gage	Right gage
0	0	0
6,069	.0005	.0005
8,092	.0010	.0010
10,115	.0010	.0010
12,138	.0015	.0015
14,161	.0025	.0025
16,184	.0040	.0040
18,207	.0045	.0045

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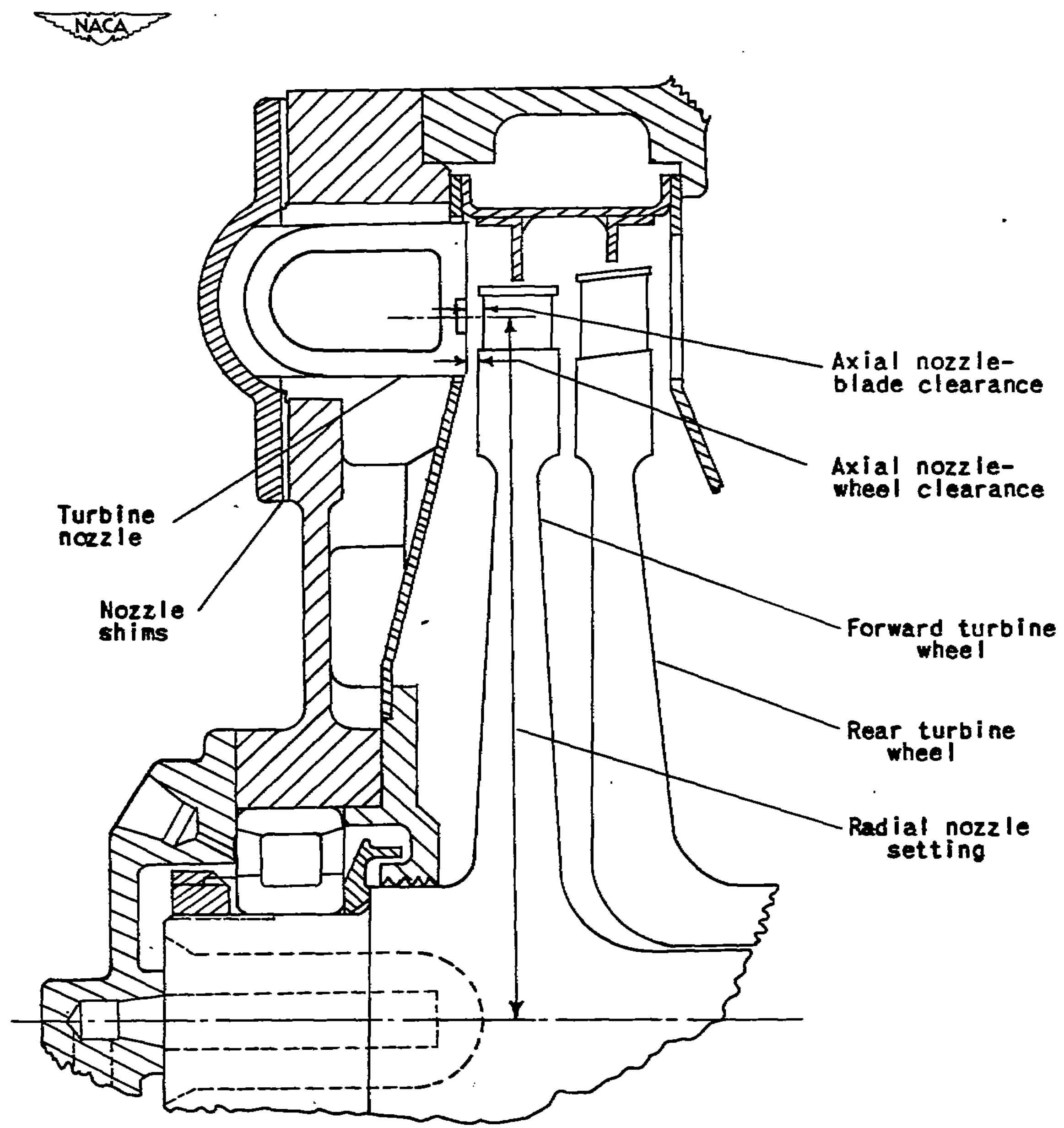
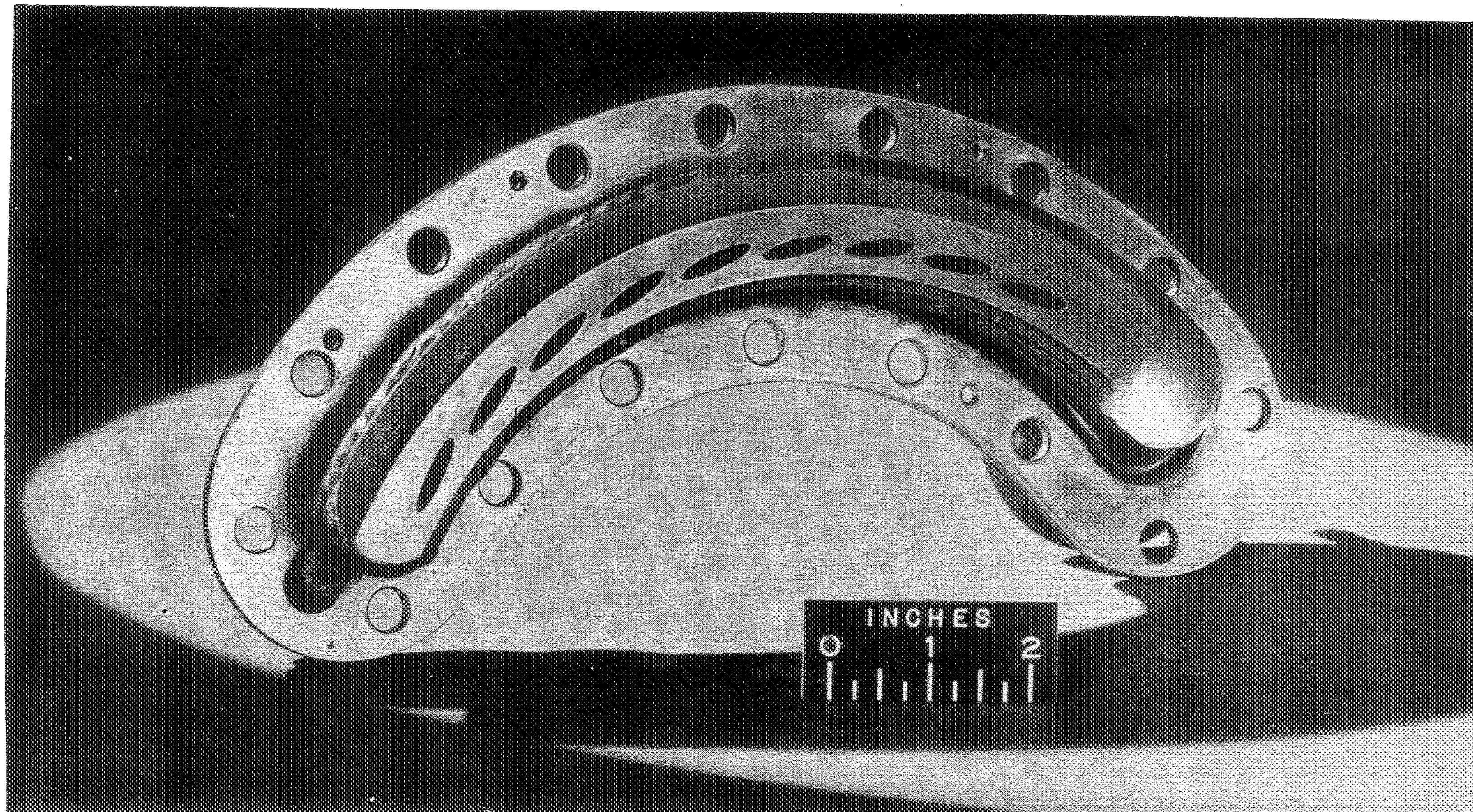


Figure 1. - Sketch of nozzle-and-turbine-wheel assembly for Mark 25 torpedo power plant.



(a) Reamed nozzle E.

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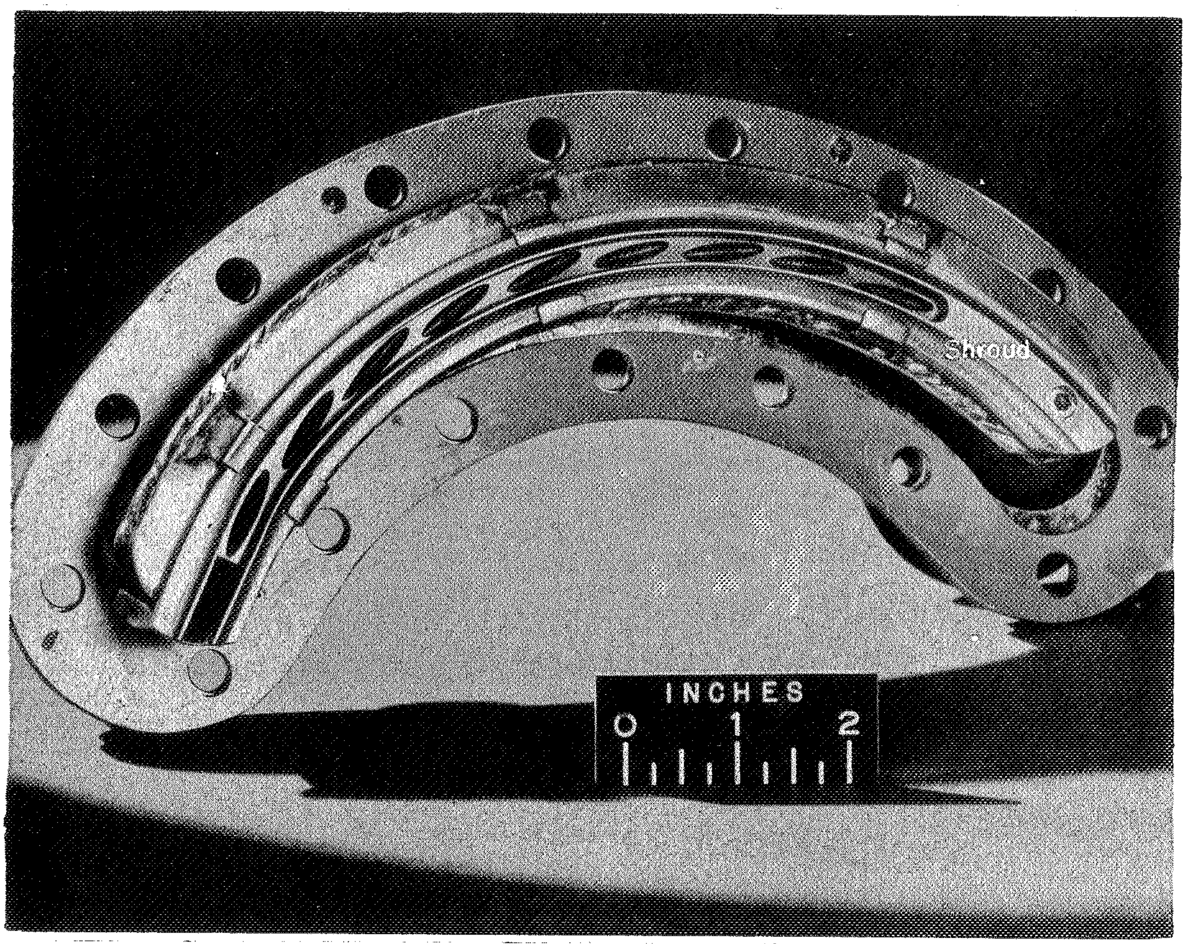
C-15949
10-2-46

Figure 2. - Outlet face of various nozzles designed for turbine of Mark 25 torpedo power plant.



(b) Reamed, shrouded nozzle G.

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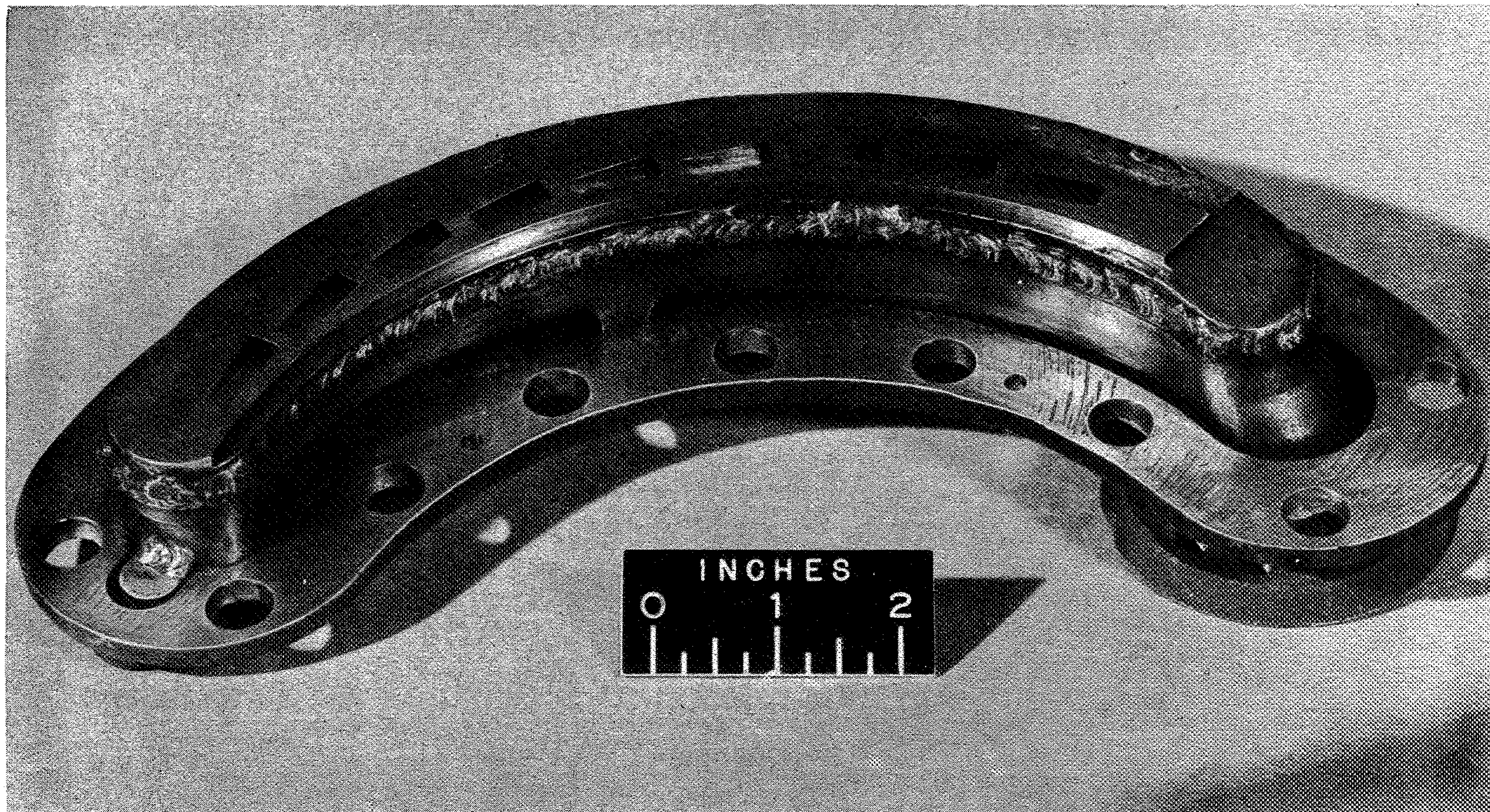
10-2-46

Figure 2. - Continued. Outlet face of various nozzles designed for turbine of Mark 25 torpedo power plant.



(c) Cast nozzle H.

NACA
C-18384
4-23-47

Figure 2. - Concluded. Outlet face of various nozzles designed for turbine of Mark 25 torpedo power plant.

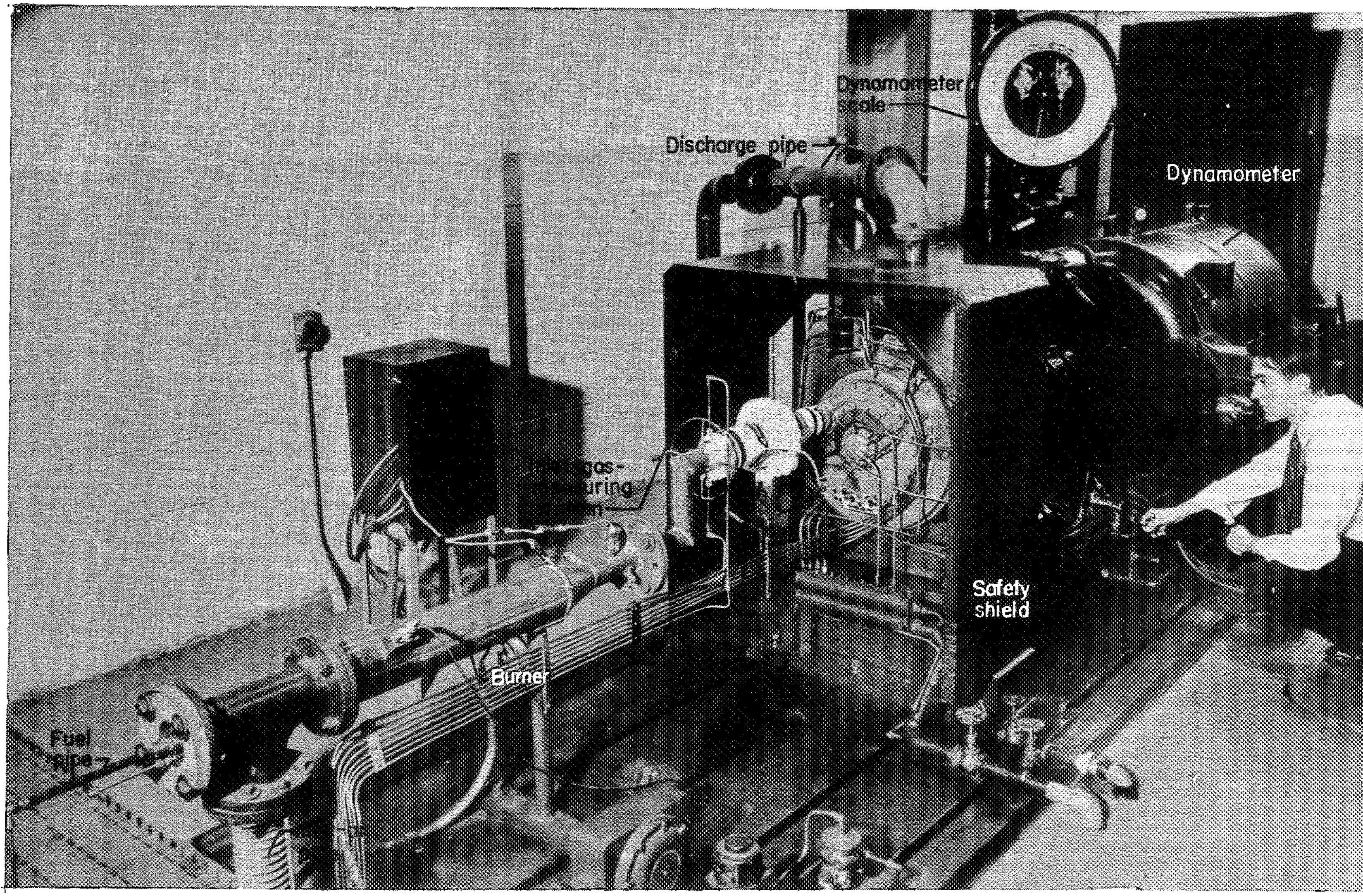


Figure 3. - Setup for investigation of turbine of Mark 25 torpedo power plant.

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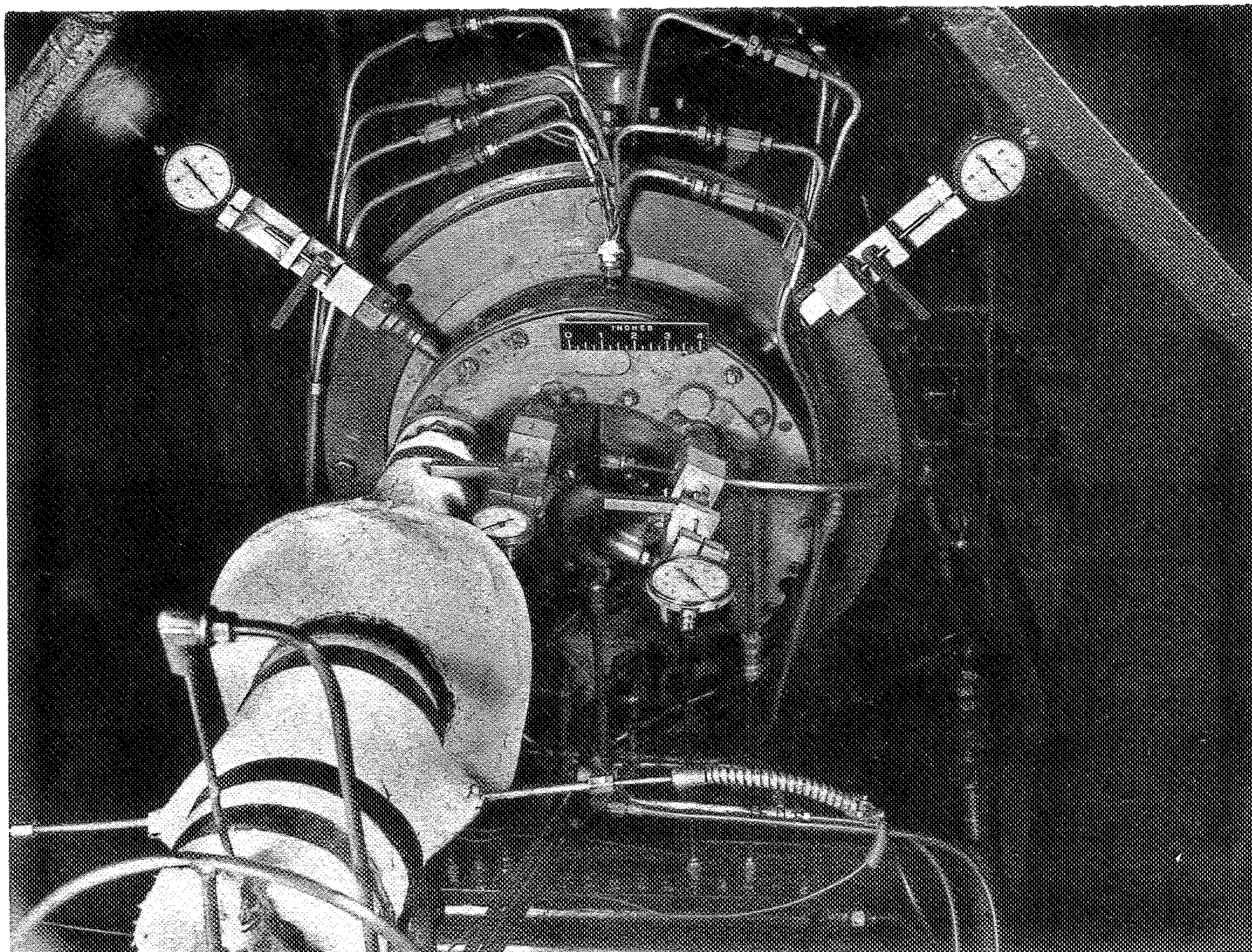
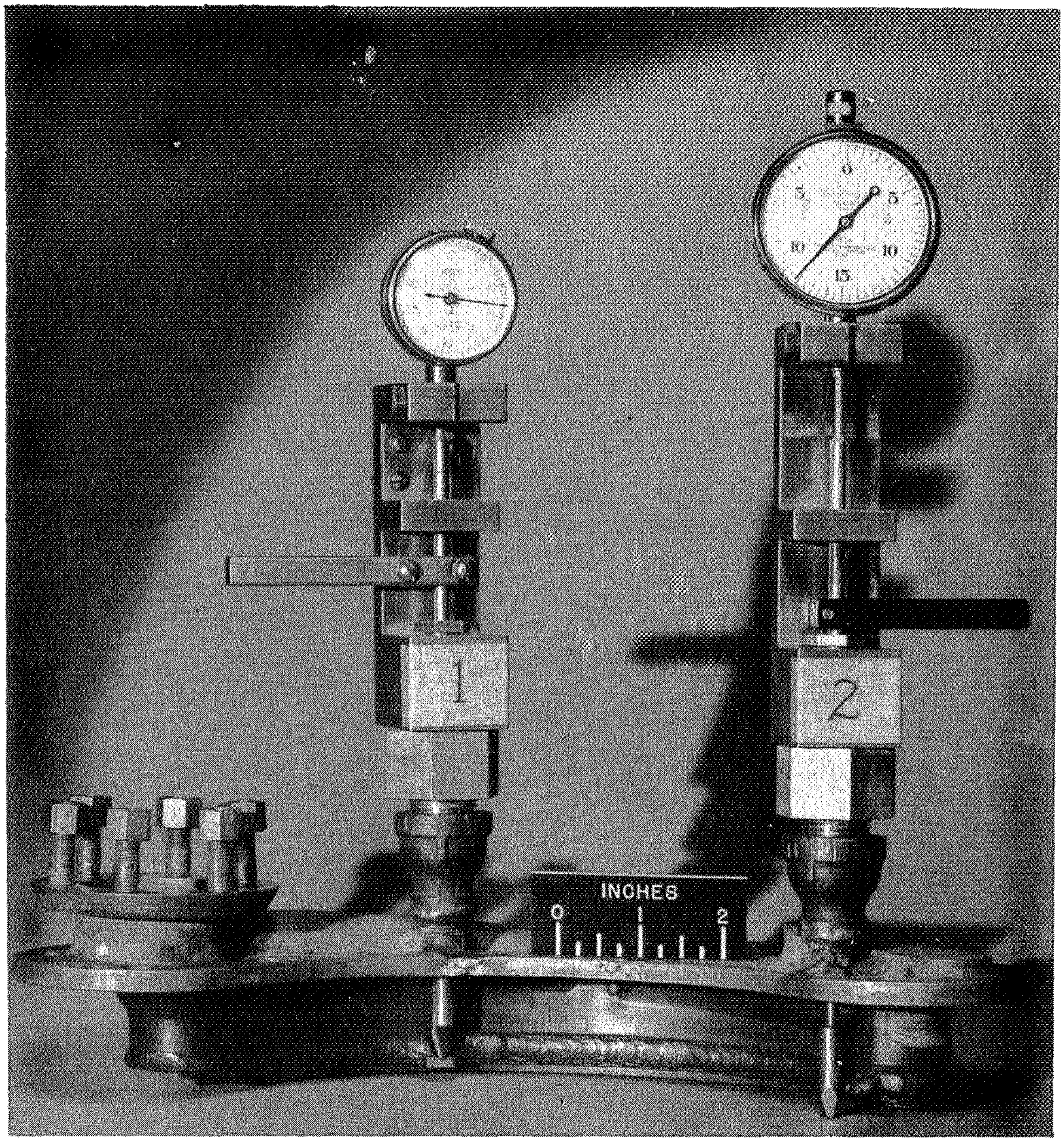
8-2-46



C-17837
2-7-47

Figure 4. - Clearance indicators installed on turbine unit to measure changes in radial nozzle setting and axial nozzle-wheel clearance.



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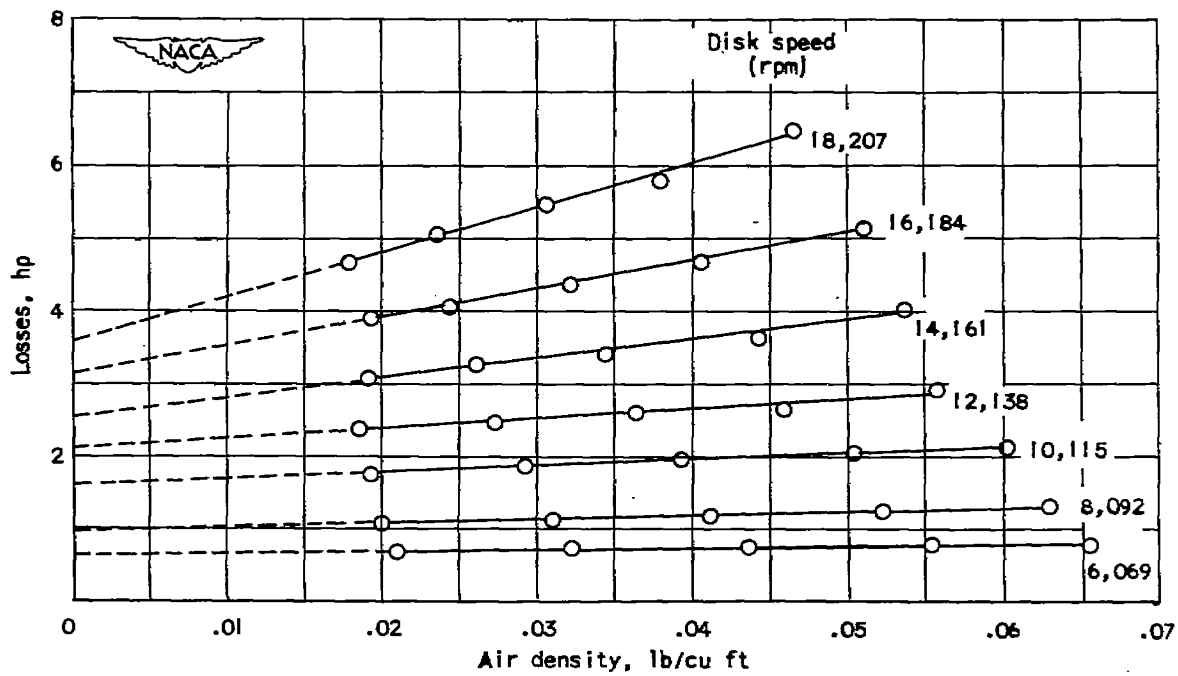
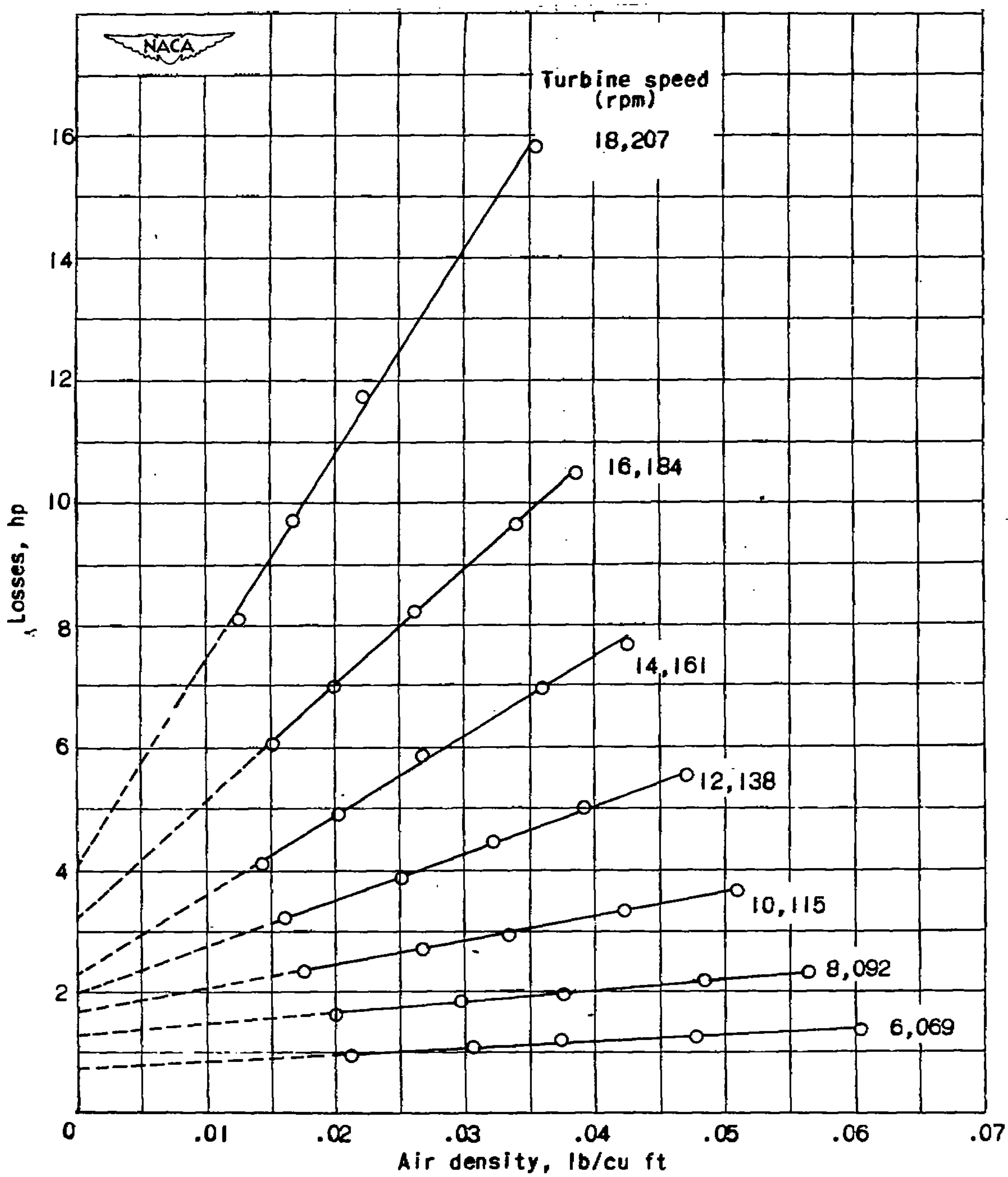
C- 18047
3-4-47

Figure 5. - Axial-clearance indicators installed on nozzle.



(a) Disk unit.

Figure 6. - Variation of windage and mechanical losses with air density and speed.



(b) Turbine wheels with blades.

Figure 6. - Concluded. Variation of windage and mechanical losses with air density and speed.

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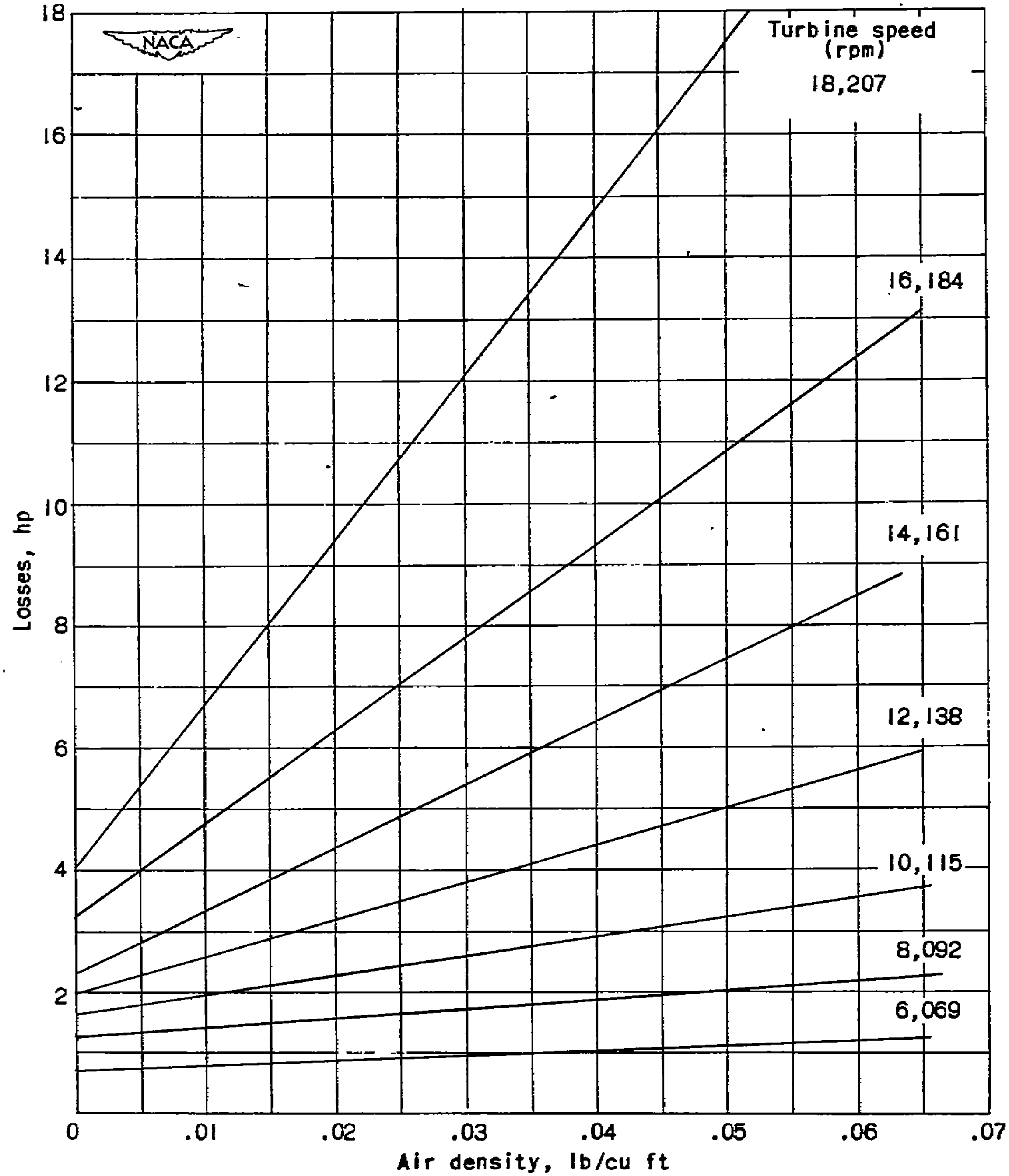
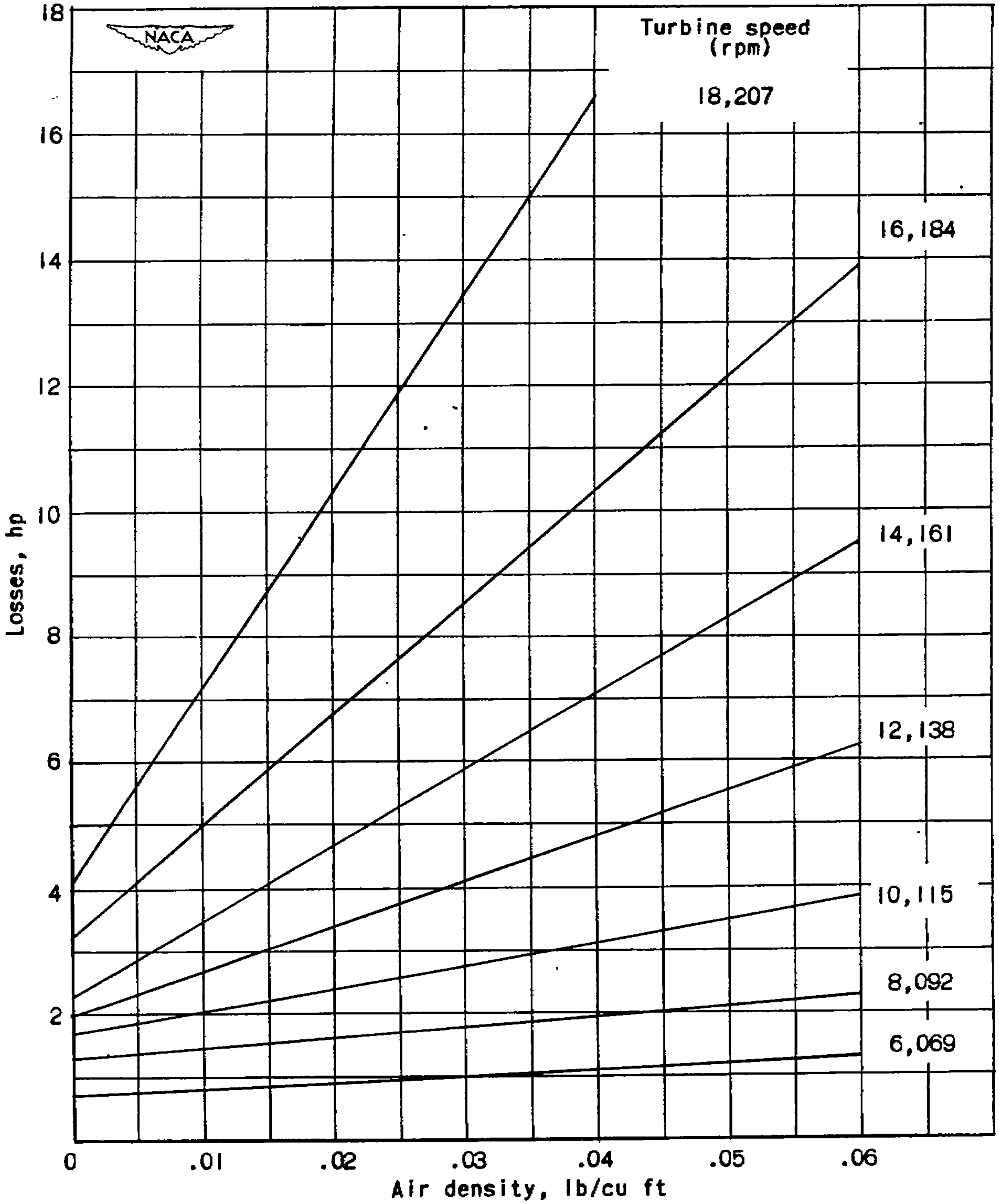


(a) Disk loss plus three-fourths blade loss for nine-port nozzle.

Figure 7. - Variation of calculated windage and mechanical losses with air density and turbine speed for partial gas admission. (Data from fig. 6.)



(b) Disk loss plus eleven-twelfths blade loss for three-port nozzle.

Figure 7. - Concluded. Variation of calculated windage and mechanical losses with air density and turbine speed for partial gas admission. (Data from fig. 6.)

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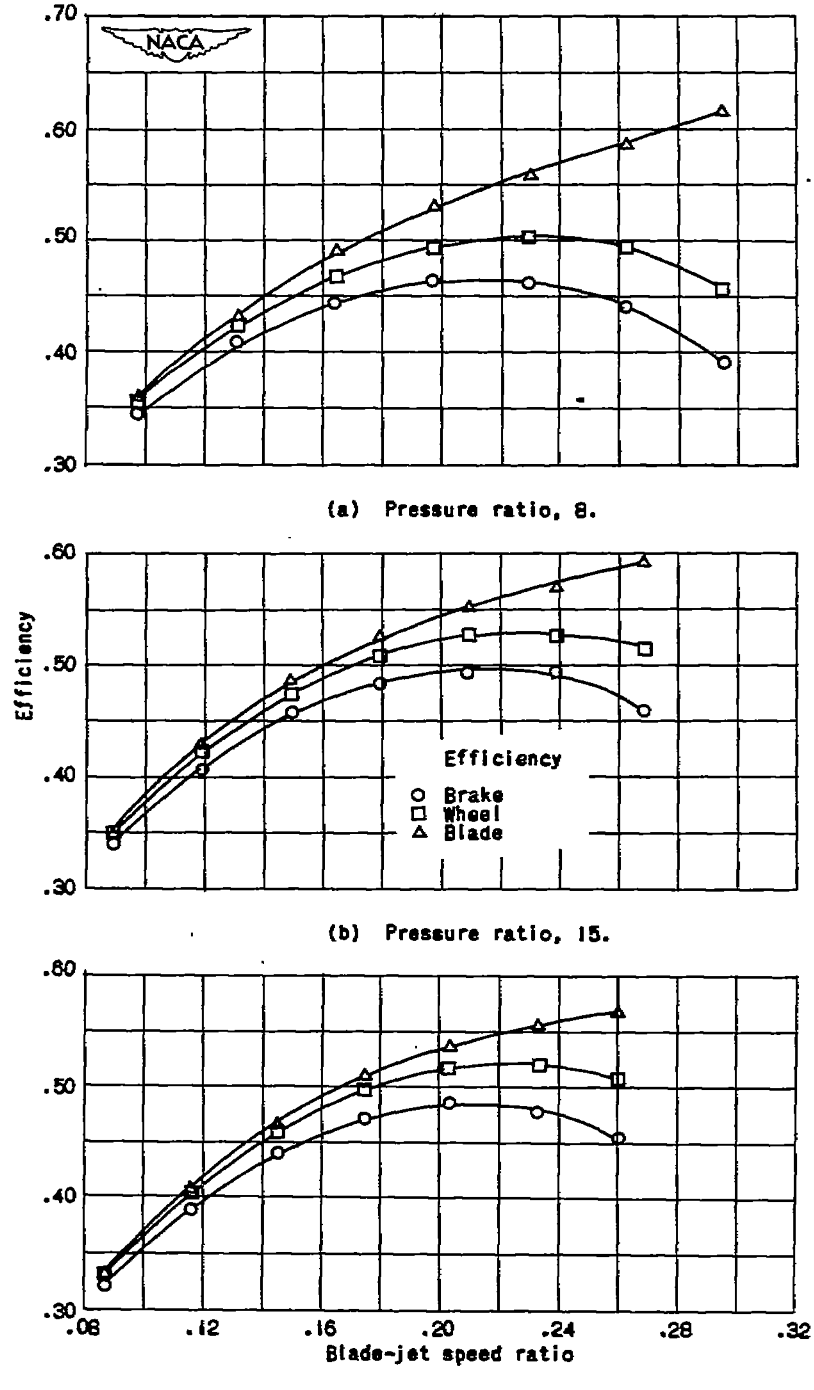
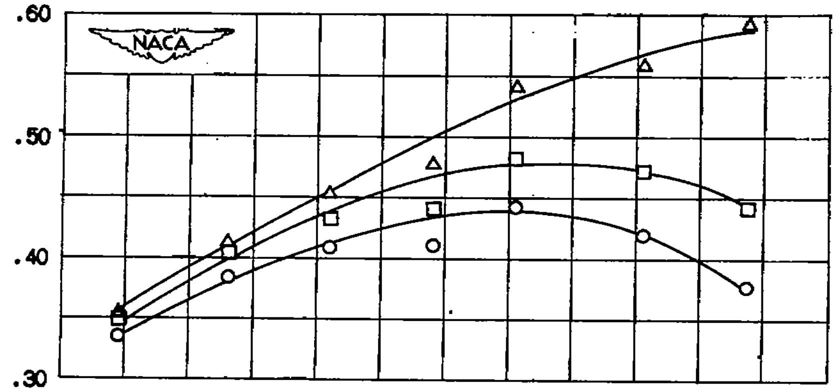


(a) Pressure ratio, 8.

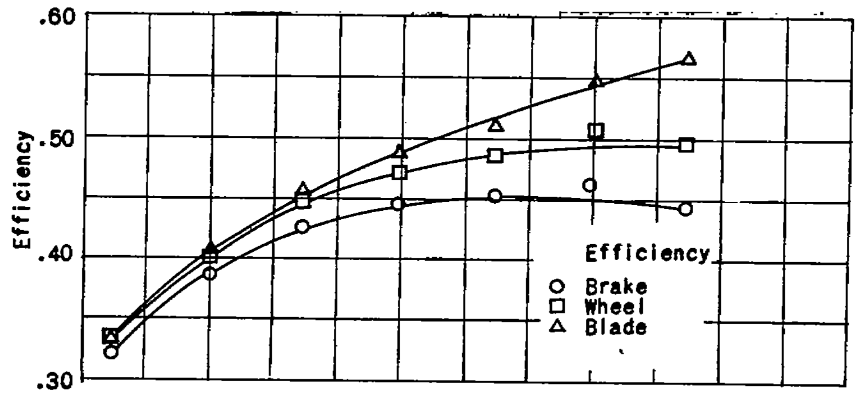
(b) Pressure ratio, 15.

(c) Pressure ratio, 20.

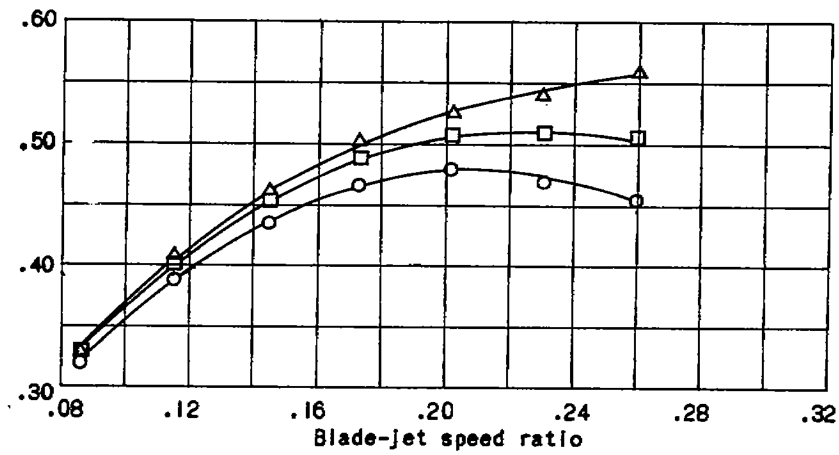
Figure 8. - Variation of power-plant component efficiencies with blade-jet speed ratio for nozzle A at three pressure ratios. Inlet-gas pressure, 95 pounds per square inch gage; inlet-gas temperature, 1000° F; axial nozzle-wheel running clearance, 0.030 inch.



(a) Pressure ratio, 8.



(b) Pressure ratio, 15.



(c) Pressure ratio, 20.

Figure 9. - Variation of power-plant component efficiencies with blade-jet speed ratio for nozzle E at three pressure ratios. Inlet-gas pressure, 95 pounds per square inch gage; inlet-gas temperature, 1000° F; axial nozzle-wheel running clearance, 0.030 inch.

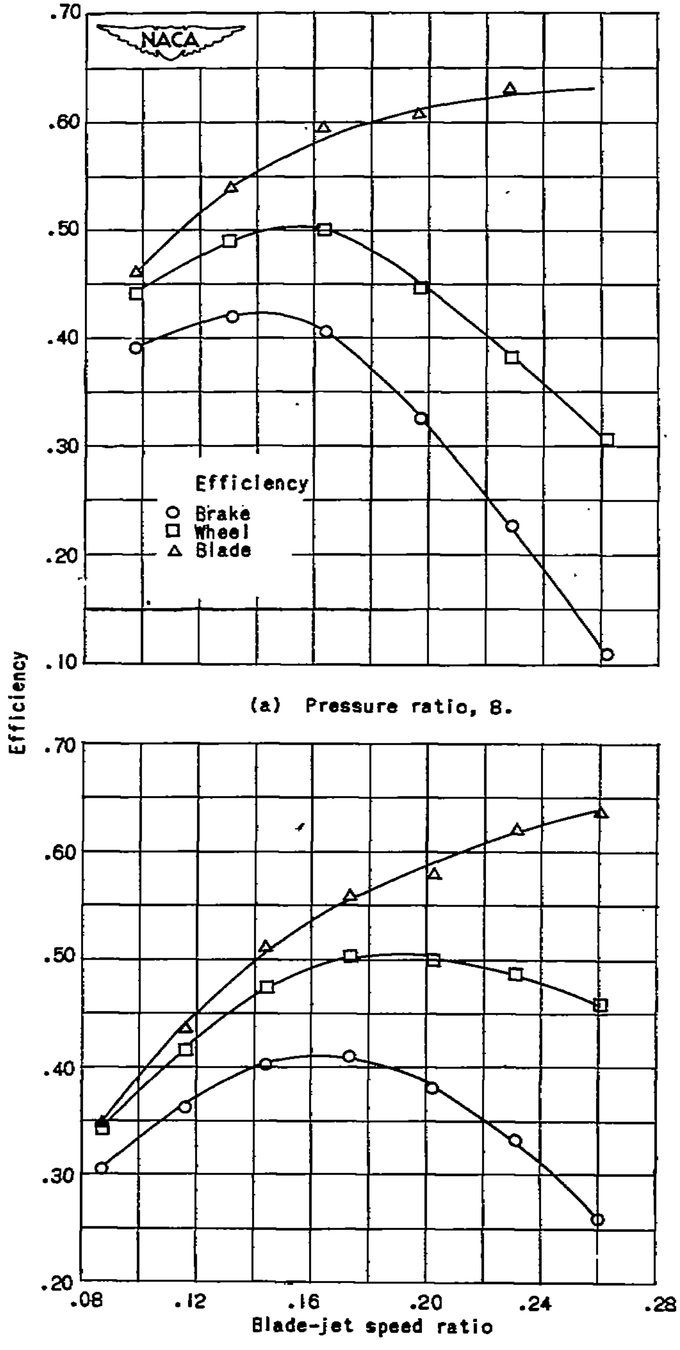


Figure 10. - Variation of power-plant component efficiencies with blade-jet speed ratio for nozzle F at two pressure ratios. Inlet-gas pressure, 95 pounds per square inch gage; inlet-gas temperature, 1000° F; axial nozzle-wheel running clearance, 0.030 inch.

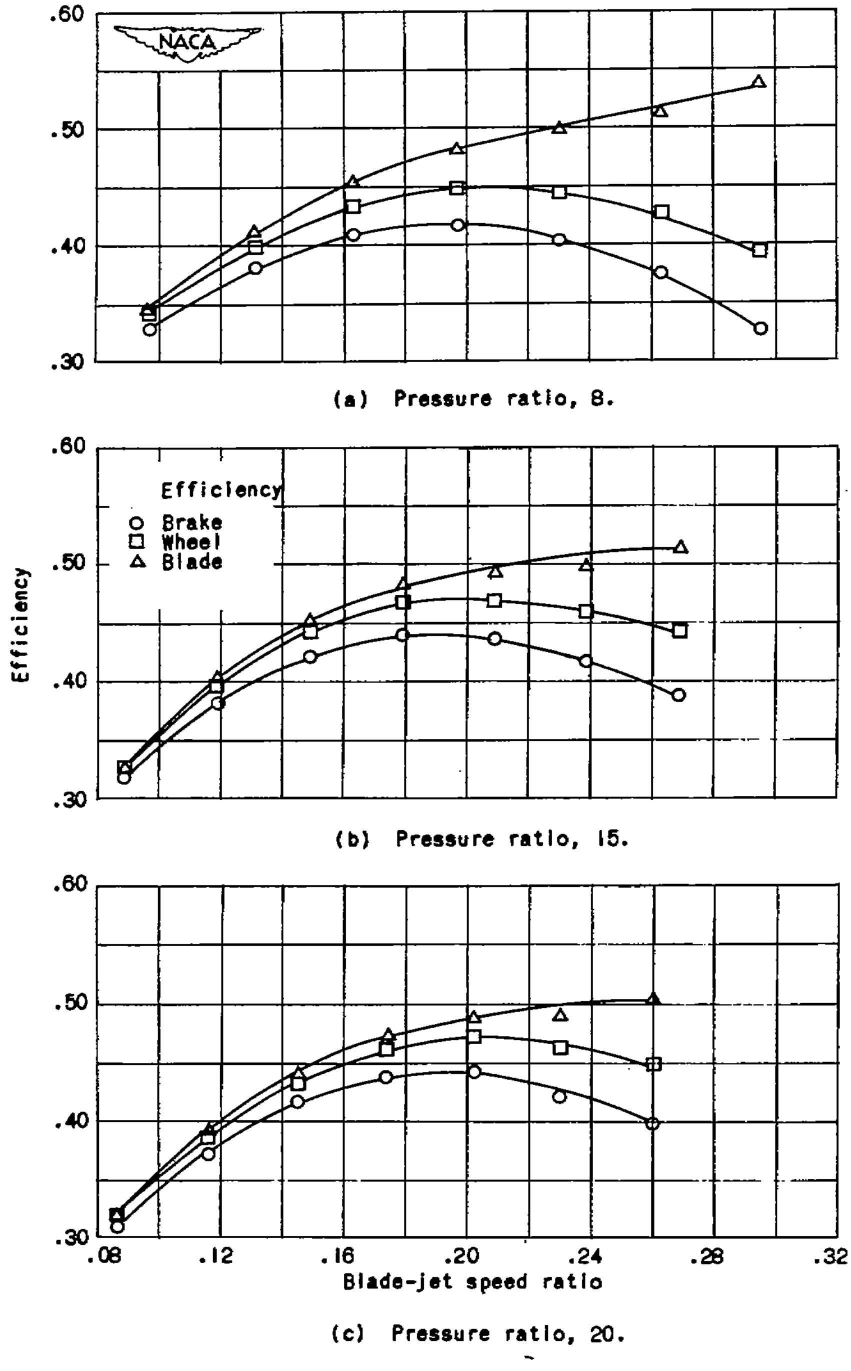
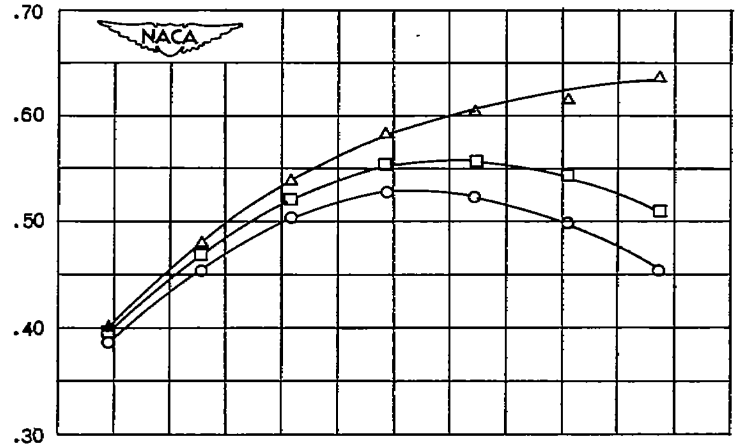
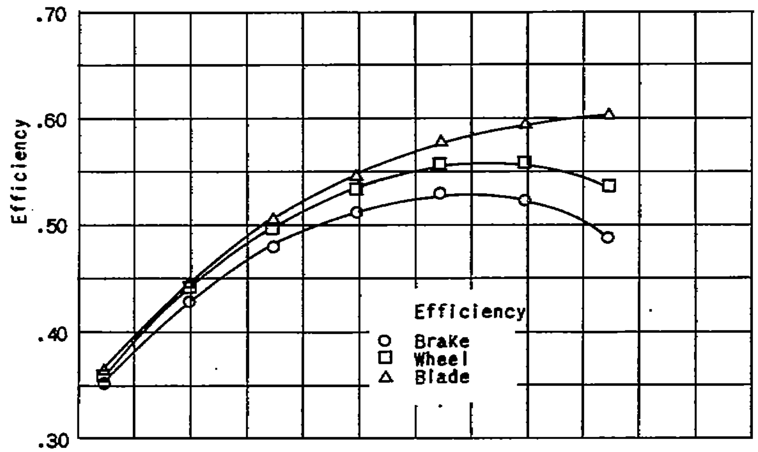


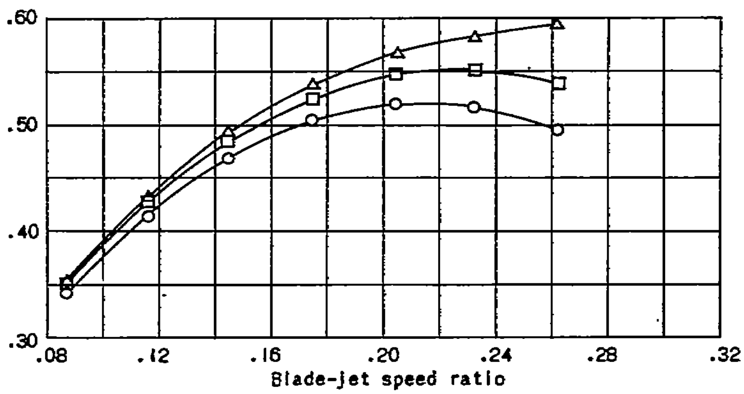
Figure 11. - Variation of power-plant component efficiencies with blade-jet speed ratio for nozzle G at three pressure ratios. Inlet-gas pressure, 95 pounds per square inch gage; inlet-gas temperature, 1000° F; axial nozzle-wheel running clearance, 0.030 inch.



(a) Pressure ratio, 8.

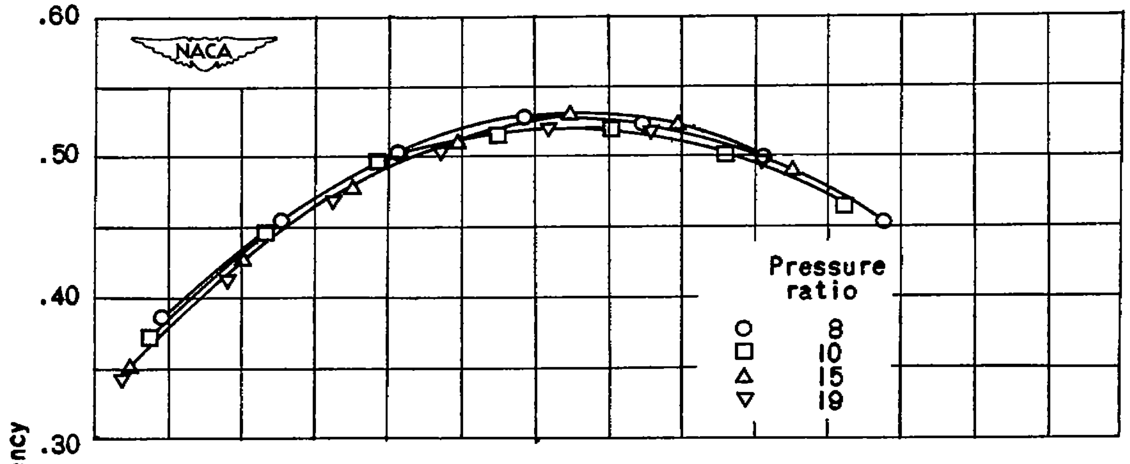


(b) Pressure ratio, 15.

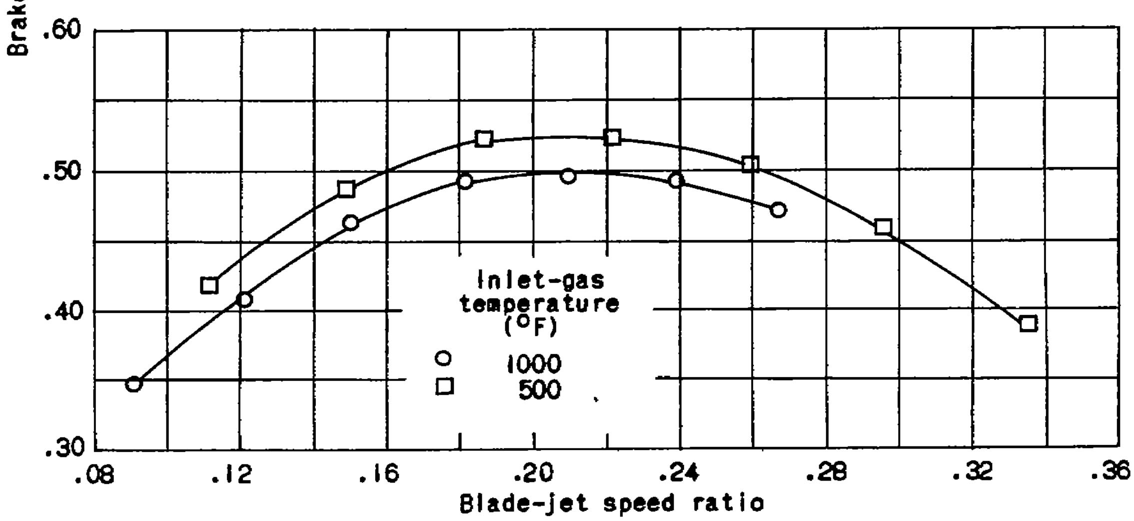


(c) Pressure ratio, 19.

Figure 12. - Variation of power-plant component efficiencies with blade-jet speed ratio for nozzle H at three pressure ratios. Inlet-gas pressure, 95 pounds per square inch gage; inlet-gas temperature, 1000° F; axial nozzle-wheel running clearance, 0.030 inch.

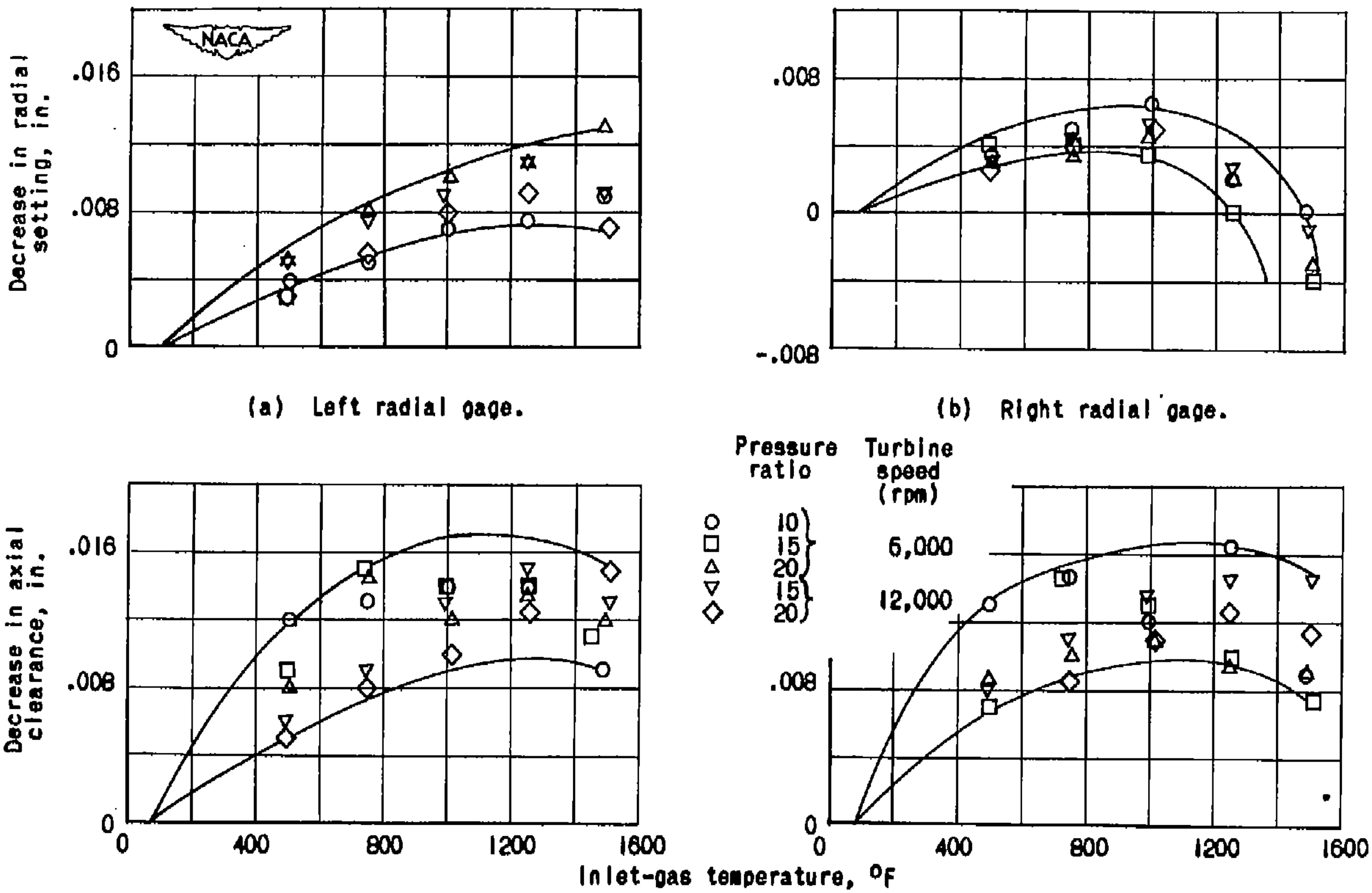


(a) Effect of pressure ratio with nozzle H at 0.030-inch axial nozzle-wheel clearance and inlet-gas temperature of 1000° F.



(b) Effect of inlet-gas temperature with nozzle A at 0.060-inch axial nozzle-wheel clearance and pressure ratio of 15.

Figure 13. - Variation of turbine brake efficiency with pressure ratio and inlet-gas temperature at inlet-gas pressure of 95 pounds per square inch gage.



(a) Left radial gage.

(b) Right radial gage.

(c) Left axial gage.

(d) Right axial gage.

Figure 14. - Changes in radial setting and axial clearance from cold clearances with inlet-gas temperature at inlet-gas pressure of 95 pounds per square inch gage.

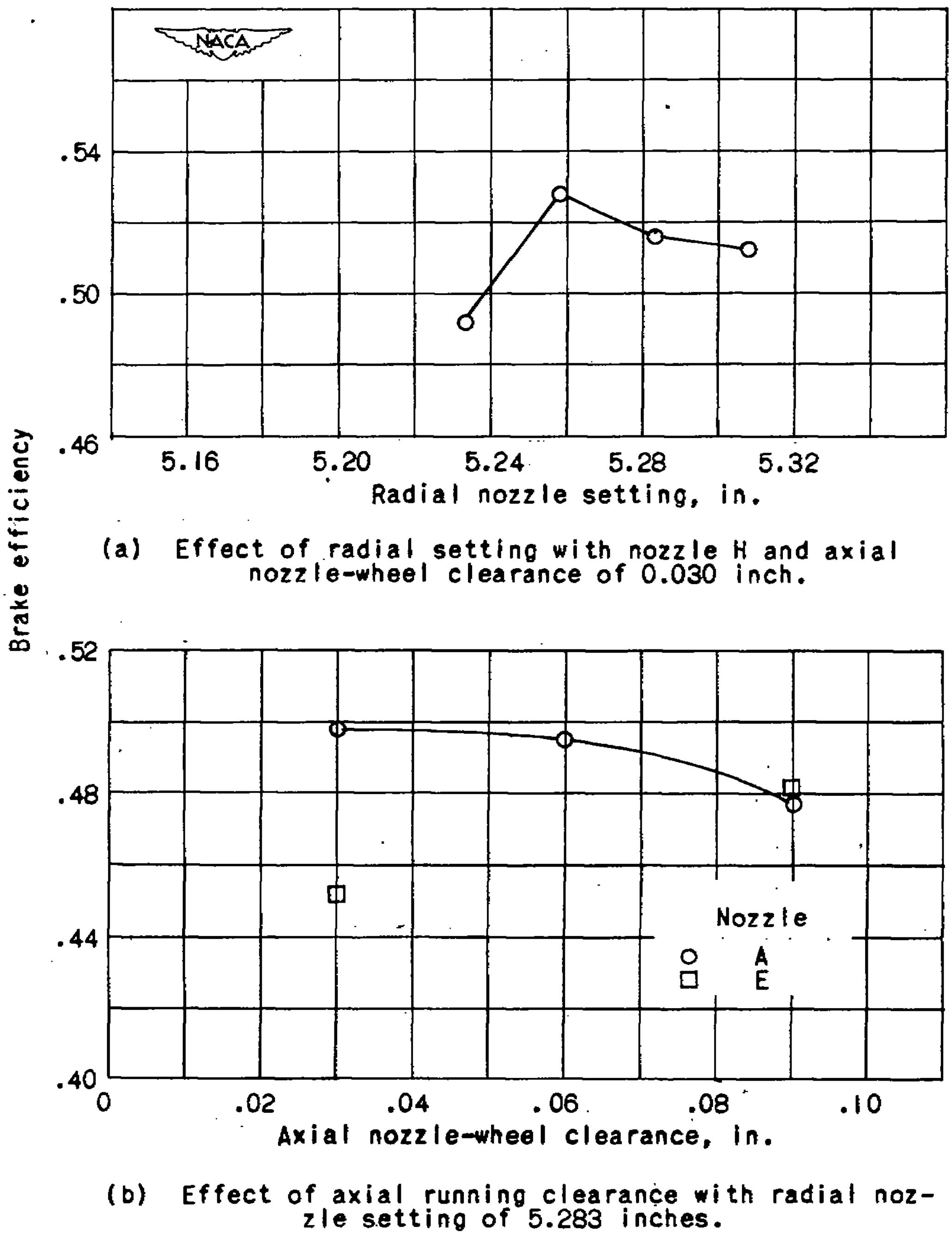


Figure 15. - Variation of turbine brake efficiency with radial nozzle setting and axial clearance. Inlet-gas pressure of 95 pounds per square inch gage; inlet-gas temperature of 1000° F; blade-jet speed ratio, 0.21; pressure ratio, 15.

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